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THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

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The Fifteenth Annual Meeting of the Society

An unequivocal success, the 1920 meeting of the Society was declared by the members in general to constitute the most valuable and edifying group of sessions that has ever been scheduled in any half-year of the organization. The intention to make the professional features of broad practical value to all engaged in the various automotive engineering fields, was excellently carried out. The interest was closely, even tensely, sustained, particularly on the morning and afternoon of Jan. 8 when the papers and discussion on fuel and research were presented.

The novelty of showing from the speakers' platform indicator diagrams from an internal-combustion engine in operation was much appreciated. The engine was run on various fuels and other substances not generally so employed. Moreover, the meeting was the occasion of the first public demonstration of recently developed high-speed indicators, indicator cards being projected on a screen to make plain the pressure variation in the engine with changing conditions of fuel. Thomas Midgley, Jr., narrated the steps that have been taken in progress in engine indicator work and described the apparatus produced by the company with which he is associated. Dr. H. C. Dickinson demonstrated the scientific indicator developed recently at the National Bureau of Standards. Illuminating papers on engines, aluminum pistons, steam systems, body design and spring suspension were also presented.

The meeting resulted in a better understanding of the elements underlying the combustion of engine fuels. An exposition was given of laboratory methods employed in measuring the rate of flame propagation in engine cylinders and the relation which this bears to engine performance. The subject of mixed fuels was treated briefly. Prof. O. C. Berry of Purdue University reported on tests he had made of wet and dry mixtures. Major G. E. A. Hallett gave information concerning supercharging aeronautic engines. G. A. Kramer, who has been conducting lubricating research work at the Bureau of Standards, discussed the effect on oils in the crankcase of dilution by fuel, and outlined methods of preventing and minimizing this effect.

In his scholarly address President Charles M. Manly took as his theme technological knowledge as the basis of all industrial development, ancient and modern, and pointed to many fundamental facts well worth the study of the members. This address, as well as the papers presented at the meeting that have not been published or sent to the members in separate pamphlet form, will appear in early issues of THE JOURNAL.

Financial and Membership Reports

The Treasurer's Report for the fiscal year ended Sept. 30, 1919, showed a gratifying condition, the net income for the period being $49,132.27. The income for the year, including dues, initiation fees and publication, advertising and miscellaneous sales, amounted to $215,031.07. The operating cost for the year, aside from contingencies, bad debts and depreciation, aggregating $6,281.88, was $159,616.92.

It was reported on behalf of the Membership Committee that in the last calendar year 964 applications for membership had been received, excluding affiliate member representatives and enrolled students, and that of these 830 or about 85 per cent had been approved and had qualified as members. In 1919, taking into account resignations, members dropped for non-payment of dues and the like, the net gain in the Society membership was 688, or approximately 19 per cent. It is considered that the new membership is of a sound nature and constituted of men of a high average of ability and standing.

On Dec. 1, 1919, the total membership of the Society was 4358. Including affiliate member representatives and enrolled students, but not Section Associates, there were on the rolls of the Society at that date 4549 names.

Constitutional Amendments

The principal new business that was brought up at the business session of the meeting was in the form of proposed constitutional amendments, duly submitted in writing and seconded in the manner provided in the Society Constitution as to amendment thereof. The proposed amendments are to be mailed to each voting member of the Society 60 days prior to the next meeting of the Society, that is the Summer Meeting, the time of which has not yet been set.

One of the amendments submitted refers to the qualifying age for Member grade, which it is proposed shall be changed from 26 to 32 years. The same thing applies to Service Member and Foreign Member grades, eligibility to these grades of membership being based in the Constitution of the Society on the same qualifications as

those for Member grade. In turn persons under 32 years of age would be eligible to Junior grade but upon reaching the age of 32 would be transferred to Member or Associate grade. It was further proposed that the annual dues for Junior members over 28 should be the same as for Members and Associates, that is $15, instead of $5, the amount of the annual dues for all Junior members at the present time.

The other proposed amendment refers to eligibility for and service on the Council of the Society. Some of the new matter proposed reads as follows, it being the intention that it shall be inserted in paragraph 29 of the Society Constitution:

> "The Council shall be chosen from among members who have been in the membership not less than two years and who have demonstrated by effective work as members of Society or Section Committees or otherwise both willingness and fitness to devote time and energy to the accomplishment of Society work. The five Second Vice-presidents shall preferably be chosen to represent respectively motor-car, aviation, tractor, marine and stationary internal-combustion engineering, but when men of suitable standing, experience and proved desire to serve the Society are not, in the opinion of the Nominating Committee, available from each respective field mentioned, two or more from other fields shall be chosen."

It was further proposed that paragraph 34 of the Constitution, with regard to vacating offices, shall be amended to read as follows:

> "The Council shall declare any elective office vacant in case its incumbent is absent from four successive Council meetings, or in case of other failure to perform the duties of his office, unless by unanimous vote of the other members of the Council such action is deemed as being not in the best interests of the Society. The Council shall thereupon appoint a Member or Honorary Member to fill the vacancy until the next Annual Meeting. The said appointment shall not render the appointee ineligible to election to any office."

The Standards Committee reported for approval at the meeting fifty-one recommendations of Standards and Recommended Practices. The number of Divisions of the Committee reporting was fifteen, as compared with ten and seven at the two last previous meetings of the Society respectively. At the June, 1919, meeting of the Society thirty-three subjects were reported and at the February, 1919, meeting thirty-one subjects.

The 1919 Standards Committee consisted of eighteen Divisions. The total number of S. A. E. Standards and Recommended Practices that had been adopted prior to the 1920 Annual Meeting was 262. The Standards Committee has nearly 200 items of unfinished work before it at this time. The specific action taken with regard to standard recommendations at its January, 1920, meeting will be reported fully in THE JOURNAL.

ELECTION OF OFFICERS

R. J. Broege, C. B. Veal and F. P. Gilligan were appointed as tellers of election of officers to serve during this administrative year and of Councilors to serve during 1920 and 1921.

They reported that 759 valid and 19 invalid ballots had been cast, the total count on election being as follows:

For President

(To serve for one year)

J. G. Vincent	732
H. M. Crane	3
Joseph Bijur	1
J. G. Utz	1
C. E. Lucke	1

For First Vice-President

(To serve for one year)

J. G. Utz	744
David Beecroft	2
J. V. Whitbeck	1

For Second Vice-President
Representing Motor Car Engineering

(To serve for one year)

W. G. Wall	742
G. A. Weidely	1

For Second Vice-President
Representing Aviation Engineering

(To serve for one year)

G. L. Martin	739
W. B. Stout	1
J. A. Steinmetz	1

For Second Vice-President
Representing Tractor Engineering

(To serve for one year)

H. C. Buffington	743
R. O. Hendrickson	1

For Second Vice-President
Representing Marine Engineering

(To serve for one year)

C. A. Criqui	743

For Second Vice-President
Representing Stationary Internal-Combustion Engineering

(To serve for one year)

L. M. Ward	742

For Councilors

(To serve for two years)

N. B. Pope	752
A. W. Scarratt	751
F. M. Germane	748

For Treasurer

(To serve for one year)

C. B. Whittlesey	753

Those receiving the largely preponderating numbers of votes were the nominees of the regular nominating committee, no other nominations having been made, and were elected. President Vincent has entered actively on his duties as the administrative head of the Society, as

is indicated elsewhere in this issue of THE JOURNAL.

There are fifteen voting members of the Council. Eleven have been newly elected as stated above. The other four are Past-president Manly and Councilors David Fergusson, E. A. Johnston and E. A. De Waters.

Social Features

The Carnival held on the evening of Jan. 7 measured up to and beyond the great claims made in the announcement of it. It was what it was planned primarily to be, a gathering at which the members could and would become better acquainted. The members certainly did "mix" and under novel and most diverting circumstances. All had to salute the special traffic cop before being admitted to the specially prepared S. A. E. scenario at the Hotel Astor, and the acquirement of phony money was essential to the taking part in any exchange or barter, which in turn it was impossible to resist amid the gayest and smartest surroundings, not forgetting the gambling hell, the crystal gazer, the nigger heads, the street organ, the Sis, or the staid Punch and Judy show.

The Carnival was indeed a beautiful and a most enjoyable occasion. The man who drew fine music from the saw will not soon be forgotten. The crowd was surely fine and the dancing floor excellent.

The members are much indebted to the Carnival Committee, and especially to Norman Bell, chairman, H. G. McComb, H. L. Spohn and A. C. Bergmann. The 1920 Annual Dinner with over 1600 seated broke all attendance records, and it was as good and satisfactory as it was large. Toastmaster John Kendrick Bangs was in great form. President Manly made some very pertinent remarks on the status and self-respect of the engineer in the automotive industry. President-elect Vincent told in a convincing way of the program of the Society for this year. Past-president Kettering entertained the members greatly with his words of common sense and satire as usual. President Charles E. Mitchell, of the National City Co. of New York City, gave a forceful address in which he explained the banker's view of thrift and made a not too optimistic statement as to financial conditions in the immediate future.

The Meetings Committee and the others who took part in the preparation for the whole meeting did excellently well and fully deserve the thanks so heartily accorded them.

SOCIETY ADMINISTRATIVE COMMITTEES

PRESIDENT J. G. VINCENT has acted very promptly in the matter of appointing the members who are to serve this year on the Administrative Committees of the Society. All of the Administrative Committees are appointed annually except the Constitution Committee, which consists of three members, each having terms of office of different length and one member thereof being appointed each year. The chairman of the Constitution Committee is D. L. Gallup. Edward Orton, Jr., who has been a member of the committee, will continue to serve thereon. W. G. Wall has been appointed as the new member this year.

The Finance Committee, the importance of which in the Society activities is obvious, is composed of H. M. Swetland, chairman; Christian Girl, G. H. Houston, Alfred Reeves and E. P. Chalfant.

The House Committee, which has a close relation to the Special Council Committee mentioned hereinbelow, has as its chairman H. E. Coffin, the other members being C. F. Kettering, C. M. Vought, H. M. Crane and J. M. Schoonmaker, Jr.

The Meetings Committee, which is already busily at work planning for the 1920 Summer Meeting of the Society, will have again the advantage of the service of David Beecroft as chairman. B. G. Koether, C. F. Scott, Azel Ames and B. B. Ayers will serve as members of the main Meetings Committee.

W. A. Brush, chairman of the Detroit Section of the Society, is the chairman of the Membership Committee. Additional members of this body, which supervises the membership increase work of the Society, are H. A. Coffin, A. C. Bergmann and Dent Parrett.

The Publication Committee, which has the duty of passing on the matter of including in the publications of the Society various papers presented at Society and Section Meetings, is constituted of Daniel Roesch, chairman, Alexander Klemin, Prof. G. A. Young, Prof. H. C. Sadler and H. C. Snow.

The Sections Committee is of great importance. H. R. Corse will serve as the chairman of this committee this year, being assisted by J. A. Anglada, A. D. Wilt, Jr., and C. B. Veal.

The Society has now reached a stage of prime importance and value to the automotive industry and the nation. This is very generally recognized. The foundations are now being re-enforced and supplemented for the greater work ahead. The offices of the Society now consist of an entire floor in the Engineering Societies' Building at New York City. The space occupied amounts to 4863 sq. ft. About forty people are employed.

President Vincent has appointed a special committee to formulate ways and means to secure the best results in the most enduring way in the matter of the financing, housing and establishing of the Society in engineering circles in this country and abroad. H. E. Coffin is chairman of this committee, the other members thereof being John N. Willys, C. F. Kettering, W. E. Metzger, Christian Girl and George H. Houston. President Vincent, Past-President Manly and Chairman Swetland of the Finance Committee are ex officio Members of this committee.

Standards Committee Meeting

THE reports of the Divisions of the Standards Committee on subjects completed since the Summer Meeting at Ottawa Beach in June, which were approved at the Standards Committee, Council and Society meetings on Jan. 6 and 7, are given below, together with the discussion at the meeting of the Standards Committee. Revisions of or amendments to reports as submitted by the several Divisions, which were made in the Standards Committee meeting, are printed in brackets following each subject.

These reports should be referred to when filling in the letter ballots which will be mailed to the members in time to return them to the Society office at New York City before March 8, the date set for counting the ballots. Many of the new and revised subjects are of general importance and should receive the careful consideration of all members of the Society whose experience warrants their voting.

B. B. Bachman, chairman of the Standards Committee, presided at the meeting on Jan. 6, which was in session from 10 o'clock in the morning until about 4 o'clock in the afternoon.

The reports of the Divisions as approved by the Standards Committee were reviewed and approved by the Council immediately following the Standards Committee meeting. The reports were duly presented at the morning session of the Society Meeting Jan. 7 and approved for submission to the voting members of the Society.

Ball and Roller Bearings Division

(1) *Annular Ball Bearings—Separable Type*

The standard for the separable or open type of annular ball bearings accepted by the Society in April, 1919, page 29ca, S. A. E. Handbook, Vol. I, did not include tolerances. The Division has recommended for adoption as S. A. E. Standard the following inch tolerances for the annular ball bearings of the separable type:

Bore	+ 0.0000, — 0.0005 in.
Outside diameter	+ 0.0005, — 0.0000 in.
Eccentricity	
Inner race	0.0006 in.
Outer race	0.0012 in.

(2) *Annular Ball Bearings—Extra Small Type*

The standard for the extra small type of annular ball bearings accepted by the Society in April, 1919, page 29cb, S. A. E. Handbook, Vol. I, did not include tolerances. The Division has recommended for adoption as S. A. E. Standard the following inch tolerances for annular ball bearings of the extra small type:

Bore	+ 0.0002, — 0.0004 in.
Outside diameter	+ 0.0000, — 0.0005 in.
Width of individual rings	+ 0.0000, — 0.0050 in.
Eccentricity	
Inner race	0.0006 in.
Outer race	0.0012 in.

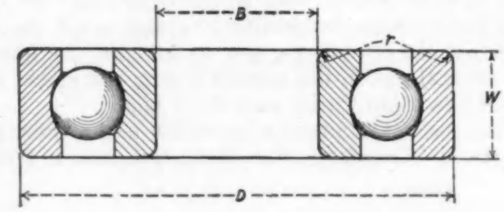
(3) *Corner Radii Tolerances*

Trouble has been experienced, particularly in aeronautical work, owing to the rounding of the edges of the outer race not being a true quarter circle. The Division has therefore recommended that all corner radii in present and future standards for ball and roller bearings be specified as minimum radii.

[Provision was made for future standards by adding "and future" after "present."]

(4) *Annular Ball Bearings—Extra Large Type—Light Series*

The Division has recommended for adoption as S. A. E. Standard the attendant series of extra large annular ball bearings. The Division is now working on tolerances for both the light and medium series.



DIMENSIONS FOR LIGHT SERIES EXTRA LARGE TYPE BEARINGS

No.	B		D		W of Individual Rings		r Corner Radius	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
224	120	4.7244	215	8.4646	42	1.6535	3	0.12
226	130	5.1181	230	9.0551	46	1.8110	3	0.12
228	140	5.5118	250	9.8425	50	1.9685	3	0.12
230	150	5.9055	270	10.6299	54	2.1260	4	0.16
232	160	6.2992	290	11.4173	58	2.2835	4	0.16
234	170	6.6929	310	12.2047	62	2.4409	4	0.16
236	180	7.0866	330	12.9921	66	2.5984	4	0.16
238	190	7.4803	350	13.7795	70	2.7559	4	0.16
240	200	7.8740	370	14.5669	74	2.9134	5	0.20
242	210	8.2677	390	15.3543	78	3.0709	5	0.20

NOTE.—For temperature of measurement and definition of eccentricity, see page 29cc, S. A. E. Handbook, Vol. I.

TOLERANCES FOR LIGHT SERIES

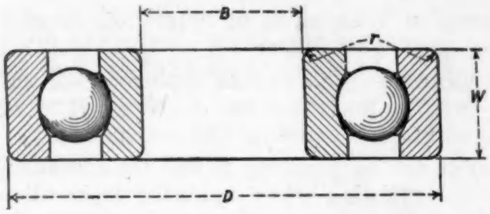
Bore	+0.0002, —0.0006 in.
Outside diameter	+0.0000, —0.0012 in.
Width of individual rings	+0.0000, —0.0050 in.
Eccentricity:	
Inner race	0.0012 in.
Outer race	0.0018 in.

(5) *Annular Ball Bearings—Extra Large Type—Medium Series*

DIMENSIONS FOR MEDIUM SERIES EXTRA LARGE TYPE BEARINGS

No.	B		D		W of Individual Rings		r Corner Radius	
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.
324	120	4.7244	260	10.2362	55	2.1654	4	0.16
326	130	5.1181	280	11.0236	59	2.3228	4	0.16
328	140	5.5118	300	11.8110	63	2.4803	4	0.16
330	150	5.9055	320	12.5984	67	2.6378	4	0.16
332	160	6.2992	340	13.3858	71	2.7953	5	0.20
334	170	6.6929	360	14.1732	75	2.9528	5	0.20
336	180	7.0866	380	14.9606	79	3.1102	5	0.20
338	190	7.4803	400	15.7480	83	3.2677	5	0.20
340	200	7.8740	420	16.5354	87	3.4252	5	0.20
342	210	8.2677	440	17.3228	89	3.5039	5	0.20

NOTE.—For temperature of measurement and definition of eccentricity, see page 29cc, S. A. E. Handbook, Vol. I.



TOLERANCES FOR MEDIUM SERIES

Bore	+0.0002, —0.0006 in.
Outside diameter	+0.0000, —0.0012 in.
Width of individual rings	+0.0000, —0.0050 in.
Eccentricity:	
Inner race	0.0012 in.
Outer race	0.0018 in.

(6) *Annular Ball Bearings—Extra-Wide Type*

The use of the extra-wide or double-row, as it is sometimes called, type of annular ball bearings is coming into prominence and the Ball and Roller Bearings Division desires to create a standard while the industry is in the formative period of development, but one company having manufactured this type of bearing extensively.

The Division has, therefore, recommended for adoption as S. A. E. Standard the following widths for the extra-wide type of annular ball bearings. The dimensions proposed conform to the widths for the present S. A. E. Standard for Roller Bearings given on pages 29d and 29e, S. A. E. Handbook, Vol. I, with the exception of No. 200 to 203, inclusive, of the Light Series, No. 300 to 303, inclusive, and 321 and 322 of the Medium Series, and all of the Heavy Series for which there are no corresponding roller bearing standards.

WIDTHS OF EXTRA-WIDE TYPE ANNULAR BALL BEARINGS

Light Series		Medium Series		Heavy Series	
No.	Width, In.	No.	Width, In.	No.	Width, In.
200	1/2	300	3/4	....	....
201	1/2	301	3/4	....	....
202	1/2	302	3/4	....	....
203	1/2	303	7/8	403	1 3/16
204	3/4	304	15/16	404	1 1/4
205	3/4	305	1	405	1 3/8
206	3/4	306	1 1/16	406	1 9/16
207	7/8	307	1 1/4	407	1 3/4
208	1	308	1 7/16	408	1 15/16
209	1	309	1 9/16	409	2 1/8
210	1	310	1 3/4	410	2 5/16
211	1 3/16	311	1 15/16	411	2 1/2
212	1 1/4	312	2 1/8	412	2 5/8
213	1 1/4	313	2 5/16	413	2 13/16
214	1 5/16	314	2 1/2	414	3 1/8
215	1 5/16	315	2 11/16	415	3 1/4
216	1 3/8	316	2 11/16	416	3 7/16
217	1 3/4	317	2 7/8	417	3 5/8
218	2	318	2 7/8	418	3 7/8
219	2 3/16	319	3 1/16	419	4 3/16
220	2 3/8	320	3 1/4	420	4 1/2
221	2 9/16	321	3 7/16	....	....
222	2 3/4	322	3 5/8	....	....

Electrical Equipment Division

(7) *Bracket Mounting for Generators*

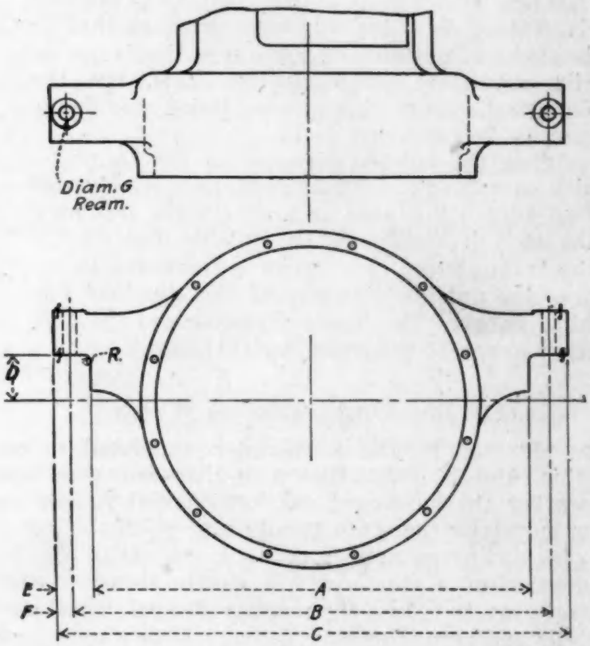
The Division has recommended that the present S. A. E. Standard for Bracket Mounting for Generators, page 36xc, S. A. E. Handbook, Vol. I, be revised so that the thread for the shaft-end will be 7/16 in.-20 S. A. E. instead of 7/16 in.-14 U. S. S., as the S. A. E. Standard thread is used in present practice instead of the U. S. Standard.

(8) *Flange Mounting for Generators*

The Division has recommended that the present S. A. E. Standard for Flange Mounting for Generators, page 36c, S. A. E. Handbook, Vol. I, be extended to specify the shaft-end as 1/2 in.-20 S. A. E. instead of 1/2 in. as at present.

Engine Division

(9) *Engine Support Arms*



DIMENSIONS FOR TRUCK AND TRACTOR ENGINE SUPPORT ARMS

No.	A	B	C	D	E	F	G	R Max	Practice
1	23 1/2	26 3/4	28 3/4	2	2 5/8	1	0.750	7/16	Truck and tractor
2	23 1/8	24 1/2	25 3/4	2 1/2	1 5/16	5/8	0.625	3/8	Truck
2T	20	21 3/4	23 1/2	3	1 3/4	7/8	0.625	1/4	Truck and tractor

In tractor practice the support arms are generally bolted to the top of the side member and in truck practice to a sub-frame or to brackets mounted inside the side member.

Engine support arms No. 1 is intended for use on flywheel housing No. 1, No. 2 on flywheel housings No. 2, 3 and 4, and No. 2T on flywheel housing No. 2.

In preparing the following proposal, data and suggestions were obtained from tractor and tractor engine manufacturers and carefully considered by a Sub-Division composed of engine and tractor transmission manufacturers. In reference to the recommendation for the No. 2T size it was considered advisable to maintain 20 in. between the side-rails because this dimension is extensively used in present practice and any increase over 20 in. would materially affect the turning radius of the tractor, which is one of the most important factors in tractor design.

As the present S. A. E. Recommended Practice for truck engine support arms, page 31b, S. A. E. Handbook, Vol. I, is now well established, the Division recommends that it be changed to S. A. E. Standard.

The proposed standard for engine support arms for the No. 1 and 2T flywheel housings for tractor practice is given elsewhere on this page together with the present

recommended practice for the No. 1 and 2 sizes for truck practice.

(10) *Tractor Flywheel Housings*

When the tractor engine and transmission case are used as a self-supported unit it is often desired to use larger capscrews than those given in the present standard. The Division has therefore recommended that a note substantially as follows be included in the present standard given on pages 31 and 31a, S. A. E. Handbook, Vol. I.

> When it is desired to use ½ in.-13 in place of ⅜ in.-16 U. S. Standard capscrews, they shall be located on the standard bolt circle, the flange inside diameter remaining the same, but the outside diameter of the transmission case flange may be increased by ½ in.
>
> When the outside diameter of the transmission case flange is not finished, as when channel iron supporting pads or brackets are cast over the edge of the flange, the outside diameter of the transmission case flange is increased ½ in. over the outside diameter of the standard finished flange. The inside diameter and the bolt circle are to be the same in either case.

(11) *Crankshaft Grinding Wheels*

Data obtained by the Subdivision appointed to consider this subject showed that with sixty-four companies representing the passenger car, commercial vehicle and tractor industries there are twenty-nine widths of crankshaft pins and seven radii in common use. It is felt that the adoption of a standard will enable abrasive wheel manufacturers to reduce the number of stock wheel sizes materially and permit quicker delivery than when wheels are made to purchasers' specifications. The proposed standard will also indicate a uniform means for specifying wheel widths, as it is stated that there is at present considerable misunderstanding regarding the extra width allowed on new wheels for dressing purposes.

The intention of the Subdivision was to standardize the crankshaft pin lengths, but the Division believes that the results desired will be accomplished better by specifying the wheel widths as now recommended for adoption.

WIDTHS AND CORNER RADII OF CRANKSHAFT GRINDING WHEELS

Nominal Widths, In.	Radii of Edges, In.
1½, 1⅝, 1¾, 1⅞, 2, 2⅛, 2¼, 2⅜, 2½, 2¾, 3, 3¼, 3½, 3¾, 4	3/32 max

This specification applies only to unfinished grinding wheels for crankshafts. Wheels are to be ordered by the purchaser to the specified nominal widths. The wheel manufacturer allows 0.029 in. additional to widths for truing. Radii of edges are specified as maximum to allow for dressing wheels to crank-pin radii.

[The footnote to the table in the original report was: "Crankshaft grinding wheel widths are specified by the nominal widths. Grinding wheel manufacturers allow 0.029 in. additional for truing wheels."]

THE DISCUSSION

H. M. Crane:—With regard to the radius of the edge of the grinding wheels, which is given as 3/32 in. maximum, I assume that is done with the understanding that the crankshaft manufacturer will modify that radius. That dimension, 3/32 in., is, of course, too small a radius to use for crankshaft fillets.

R. S. Burnett:—This radius applies to the unfinished wheel so that the manufacturer can dress it to whatever radius he wants for grinding the crankshaft.

Mr. Crane:—The footnote to the table regarding the fact that the grinding wheel manufacturers allow 0.029 in. in addition on the width for truing wheels, should be amplified to explain the reason for the 3/32 in. maximum radius, to make it clear that this also is specified to allow for truing and for reaching a radius that is properly proportioned to the crank bearing itself.

Albert Turner:—Regarding the widths for grinding wheels I believe that the gradation is too fine as proposed and that the ⅛-in. sizes should be omitted; that is, the series should progress in steps of ¼ in. instead of ⅛ in. The wheel manufacturers have standardized to a considerable extent and make wheels in 1¼-in. steps. When a wheel is ordered in ⅛-in. size it is necessary to dress it down from the next larger size, causing extra expense. If the widths were in even ¼-in. steps the wheels could be kept in stock and there would not be the additional expense of truing them down. This would also reduce the number of sizes in the table from fifteen to twelve, which I think would be ample to cover the necessary range.

The 3/32-in. radius should be specified as a minimum instead of a maximum. The wheels, when they are first made ready for shipment, are square cornered. The purpose of the radius on the corner is to make the wheel stand up in use. If the radius is too small on the corner it will break down in use, but if it is made of ample size it will stand up much longer. About ⅛ in. should be the minimum radius, but as a little variation should be allowed, a 3/32-in. minimum radius will probably be satisfactory.

Mr. Burnett:—The original proposal of the Sub-Division which prepared this report included the ⅛-in. progression of sizes as shown in the report. It was not believed wise to take out the ⅛-in. step from the narrower sizes as this would limit the number of widths too greatly. The preliminary recommendation applied to the crankshaft pins, however, rather than to the wheels. The radius originally specified by the Sub-Division applied to the crankshaft pins and was ⅛ in. up to a 3-in. pin diameter and 3/16 in. for the larger diameters. Members of the Engine Division felt, however, that the grinding wheel widths should be specified so that the crankshaft or engine builder could grind the crank-pins to any length desired, and that the radius should be a maximum to leave enough material for the user to dress the edge of the wheel to the radius of the crank-pin fillet. The dressing of the wheel is, of course, left to the user and it is intended that the dimensions in the proposal shall apply to the rough wheel as purchased from the wheel manufacturer.

Chairman B. B. Bachman:—With regard to the specifying of the radius there is probably a tendency to be a trifle confused as to whether we are speaking of the product or the tool. If we are speaking of the product, the wording should be minimum; if we are speaking of the tool, it should be maximum, because what is wanted is a wheel that will have sufficient material on the edge to allow for dressing down to a proper radius in proportion to the pin.

(*Continued on page* 57)

Bettering the Efficiency of Existing Engines

By HUGO C. GIBSON[1] (*Member*)

ANNUAL MEETING PAPER *Illustrated with* CHARTS

THE first De Dion motor tricycles were arranged to consume a fuel so volatile that it was only necessary to bubble part of the intake air through the liquid to obtain a mixture so rich in hydrocarbon vapor as to require the admixture of further air for proper combustion. This fuel was the top of the toppings from the small quantity of crude oil then being refined, and the supply was more than sufficient to meet the demand for internal-combustion engine fuel at that time. As the demand grew, these tops were insufficient and so, when the heavier distillates of a slightly higher boiling point were added to augment the supply, De Dion installed a heater pipe in the surface carbureter and passed hot exhaust gases through it to boil off those higher-boiling-point fractions which would not, without heat, evaporate rapidly enough to provide an effective mixture.

The arrival of the atomizing carbureter called a halt for a time on the developments involving the use of heat for vaporization, for with the comparatively low boiling point of that day, the minute globule coming from the spray nozzle would receive sufficient heat from the air in which it was entrained, even in the short time of transit from the nozzle to the engine valve, to evaporate it. The vaporization was aided, of course, by the vacuum in the inlet pipe, which naturally reduced the boiling point of the liquid. Those of us who used "gas wagons" in 1890 to 1895 will remember that we had little thought for carbon except such as resulted from the destructive distillation of the lubricating oil which found its way into the combustion space.

The demand for fuel became progressively greater, so that with all the low-boiling-point toppings that could be saved, the refiner was compelled to throw in some of those fractions which were formerly allowed to run in with the kerosene. Then the carbon trouble began to brew and increased so rapidly that it is now almost essential to remove carbon deposits every few weeks, especially in the case of high-compression engines.

Carbon deposits represent a waste of a large quantity of gasoline, which not only serves no useful purpose but actually results in definite costly harm to the engine, in wear due to abrasion and attrition by hard carbon particles, to say nothing of the associated harm resulting from pollution and dilution of lubricating oils by raw fuels, and such products of combustion as are washed into the crankcase with the unevaporated fuel that accompanies such a condition. An outline of the methods that have been found of value in offsetting the rising boiling points of our fuels should be helpful as a guide for the more general adoption of such means as will result in increased satisfaction and profit to all motorists, whether the profit be in pleasure and recreation or of the bankable kind attainable through commercial truck or farm machinery operation. An increase of such satisfaction means the prolongation of widespread industrial activity of paramount importance to the nation, through its automotive and petroleum industries, its third and sixth largest in value of products. Every unit of these industries would feel the pinch resulting from a hurried search for fuel to substitute for the gasoline we have thrown away, while wishing for a better fuel than we now have at the price we now pay. The problem confronting the industry threefold, involving:

(1) The physics of the carbureter, meaning the metering of a liquid into a current of air in a chemically correct quantity

(2) The physics of evaporation of hydrocarbon liquids, meaning the complete evaporation of the liquid prior to the physical mixture of its molecules with those of oxygen

(3) The physics of diffusion of gases, meaning the placing of the molecules of the vapor in close relation to their chemically appropriate molecules of oxygen, whereby inflammation may produce complete combustion with maximum temperature

These three phases accurately performed enable us to secure a maximum value from the formula

$$\text{Efficiency} = \frac{t_1 - t_2}{t_1}$$

which means that the higher the temperature of combustion at the beginning of the power stroke t_1 the less fuel we will use for the power developed. Thus, if we desire fuel efficiency, we must bend our energies to the attainment of conditions as near perfection as may be in each of the phases, for partial failure in any one deteriorates the whole effort materially.

VOLUMETRIC EFFICIENCY

One potent deterrent to the attainment of reasonable fuel efficiency has been the tendency to subordinate efficient vaporization to the attainment of maximum volumetric efficiency, meaning maximum weight of charge per stroke. The real goal to aim at is to inspire sufficient charge to develop the desired amount of power. This conception involves a little less charge per stroke, higher-speed engines with consequent increase in number of charges per minute, better chemical composition of the charge, higher initial temperatures and pressures and consequently less fuel per horsepower.

The effort of most carbureter designers has been to give high torque at high speed, where the useful range of mixture proportions is narrow. This involved the use of more nearly chemically-perfect mixture proportions for the purpose, but still the composition generally inclines toward the rich side because full torque can be maintained with less accuracy of metering than is necessary to maintain maximum fuel efficiency.

At low speeds the range of useful mixture proportions for maximum torque is much wider, the carbureter designer takes advantage of it, and since our engines are run at comparatively low speeds most of the time our

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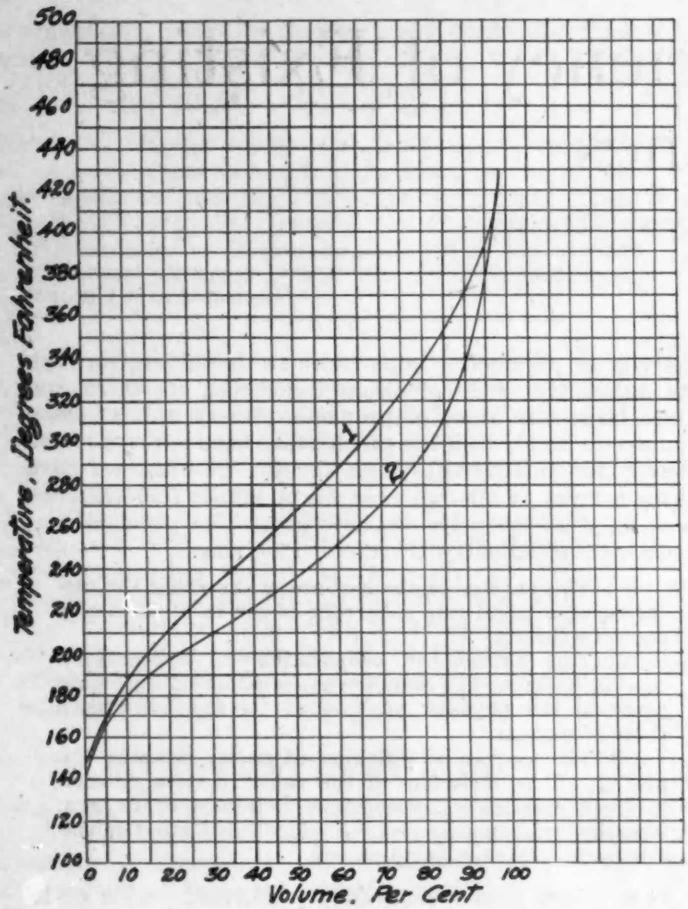


FIG. 1

greatest consumption of fuel is generally performed with the more inefficient mixtures. This tendency of the carbureter designer is another result of our insatiable demand for extreme power output from minimum engine dimensions.

Because of the desire for high volumetric efficiency per stroke, the carbureter designer tried to avoid restriction and did avoid heat because both militated against that sort of efficiency, but every such designer will agree that, with a very slight restriction, accurate metering is possible today, and now that the permissible error in metering is more nearly established, as shown by the work done at Purdue University and elsewhere, we may look for greater accuracy and efficiency in the first phase of the problem, but not if we insist on high volumetric efficiency per stroke through the use of cold mixture. Reduced sectional area of the manifold, used for the purpose of maintaining mixture speed when a cold mixture is employed, also results in reduced volumetric efficiency.

The economical mixture is always leaner than that giving maximum torque, but the difference in torque is slight and not worth the economic loss of fuel. To make up for that difference in torque we must use slightly higher engine speeds and higher gear ratios, if we wish to maintain the rapid getaway and high car-speeds to which we are now accustomed.

The carbureter designer is not to be blamed for these uneconomical uses of fuel. His customer, the engine builder, must change the nature of his demand. It will then be met as to accurate metering for economical mixture proportion. The loss of fuel from inaccurate metering may equal the amount actually necessary for the work done by the engine, or, in other words, a 50 per cent loss of fuel may occur without noticeable effect on the torque output.

Referring to the other two phases the complete evaporation of the liquid prior to chemical combustion is of vital importance to the economical combustion of fuel, for unless it is evaporated some considerable time before the spark occurs, it is impossible to comply with the requirements of the last phase. Liquid cannot burn, it must first be converted into a gas suitable for our purpose and thereafter molecularly presented to oxygen in the correct quantity. These operations take time and the length of time at our disposal is in any case minute. At 3000 r.p.m., a cold fuel molecule must leave the carbureter jet, become separated from its fellows, surrounded with oxygen, and ignited, all within 0.01 sec. This would be quick time for a photographic exposure, to say nothing of the physical transfer of the molecule along a path say 18 in. or more in length while undergoing a holocaust of physical transformation.

VOLATILITY OF FUEL

We are accustomed to think that gasoline is a volatile liquid, but if we would forget all about volatility in considering fuels we would arrive at a solution more quickly. When we desire to evaporate liquids, such as water in a boiler, we give sufficient heating surface, sufficient time and sufficient but not too high a temperature. We do not consider water a volatile liquid, yet about 80 per cent of every gallon of gasoline sold to the motorist will not boil at the boiling point of water. Thus from this standpoint volatility now becomes a negative term, while condensability would more nearly describe its important feature.

The following figures and curves show the difficulty of

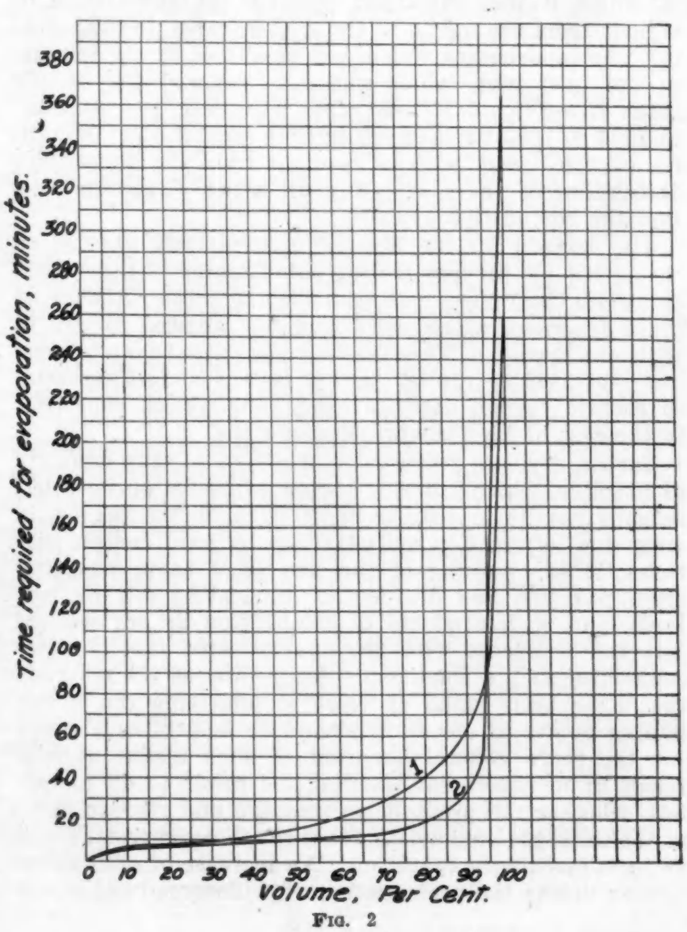


FIG. 2

evaporating gasoline unless sufficient temperature and time are applied and allowed. Two gasolines, No. 1 a Mid-West and No. 2 an Eastern product, were distilled in the ordinary way, each successive 10 per cent of distillate being collected in a separate beaker. The time required for distillations, and the initial and final boiling points, using the standard test procedure of 2 drops per sec., were, respectively

	No. 1	No. 2
Time	25 min. 51 sec.	25 min. 18 sec.
Initial Boiling Point	149 deg. fahr.	144 deg. fahr.
Final Boiling Point	419 deg. fahr.	430 deg. fahr.

Part of each of the 10 per cent cuts was then redistilled, using standard procedure, with these results

	INITIAL BOILING POINT		FINAL BOILING POINT	
Cut No.	No. 1	No. 2	No. 1	No. 2
1	111	97	320	288
2	127	118	324	293
3	147	142	334	300
4	172	162	345	306
5	198	181	358	320
6	225	199	369	333
7	255	212	392	347
8	288	235	403	367
9	313	262	424	405
10	349	315	500	531

Fig. 1 shows the normal distillation characteristics of the two gasolines, while Fig. 2 shows the time in minutes necessary to evaporate the various 10 per cent cuts at 180 deg. fahr. It is noteworthy that the final boiling point of any cut is much higher than the initial point of the next cut in the series. Figs. 3 and 4 show the distillation temperatures of each 10 per cent cut of each fuel with the distillation curve of the entire sample which is shown by the broken line superposed. From the above, it is evident that much more heat and time are necessary, for the evaporation of the high-boiling-point ends of gasoline which may be deposited on manifold walls or in cylinders, than we are accustomed to allow.

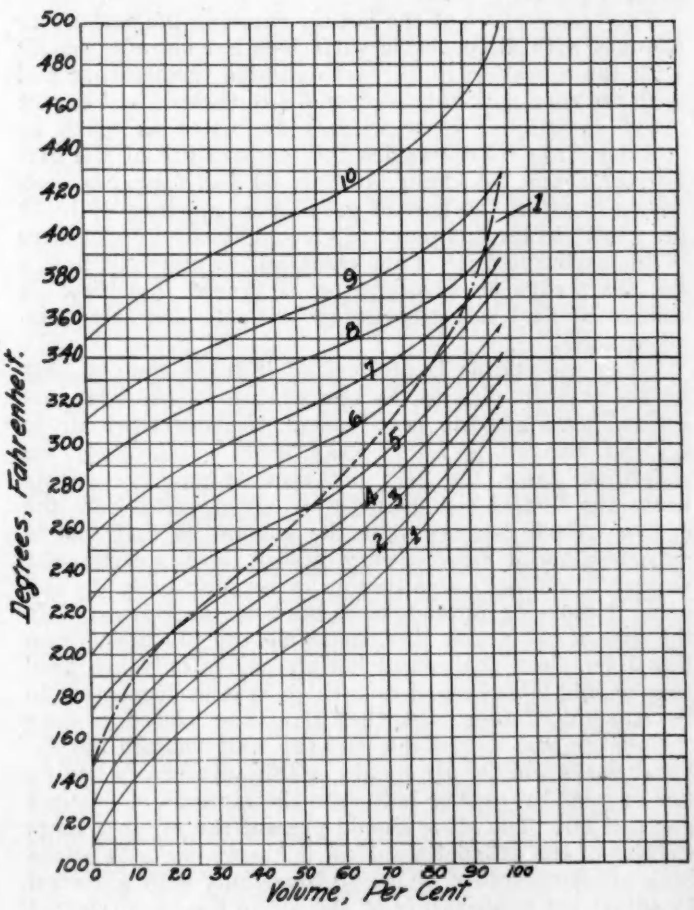


Fig. 3

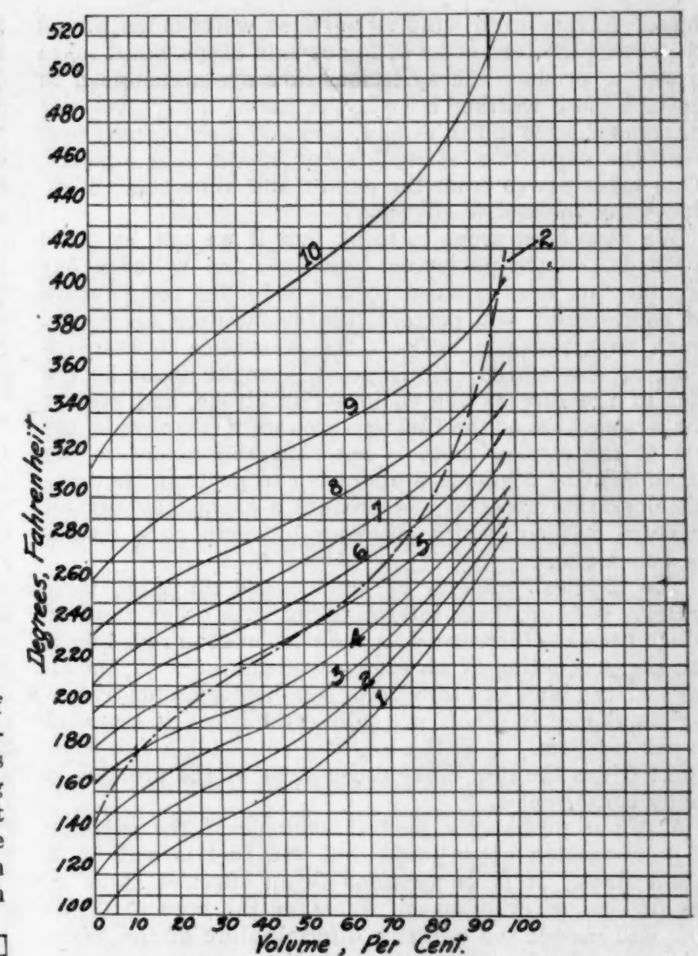


Fig. 4

Another part of each 10 per cent cut was subjected to a temperature of 180 deg. fahr. and the time necessary for complete evaporation was observed, as follows:

	No. 1		No. 2	
Cut No.	Min.	Sec.	Min.	Sec.
1	6	53	5	33
2	8	51	6	57
3	11	4	7	52
4	12	23	8	48
5	14	35	9	58
6	18	24	12	46
7	29	20	14	48
8	39	15	16	29
9	65	32	29	34
10	259	0	360	34

The carbureter puts the liquid in a condition for rapid evaporation, by breaking it up into small globules, and we ordinarily try, by proportioning and shaping our manifolds, to prevent the tendency of those globules to join with others in the formation of larger globules. In fact, we try to prevent them from following the law of nature that provides for the formation of a large drop of rain out of the assembly of a large number of fog particles. In the case of water fog in the clouds, a little local mechanical disturbance or a little drop in temperature precipitates rain. The presentation of a cold surface to a water fog results in a sheet of water. On introducing the heat of the sun, the fog disperses and the rain evapo-

rates. A very high wind so agitates water-laden air as to prevent the formation of fog or rain drops, but it takes power to produce the agitation. We are accustomed to imitate these weather laws in the use of a high velocity in manifolds, to aid in evaporation and mechanically prevent the deposition of liquids out of the cold mixture. This takes power from the engine and unnecessarily reduces the volumetric efficiency.

We have been prone to think that if we can, by these methods, induce pulverized fuel into the cylinder our troubles are over because, so it is said, the temperature of compression will perform the evaporation necessary. Such an hypothesis neglects the resistance to evaporation, or rise of the boiling point, brought about by the rise of pressure on the compression stroke, and even if the evaporation of the fuel particles were complete at the top of the stroke, there is not time subsequent to the attainment of complete evaporation to perform the mixing operation necessary to produce a chemically perfect mixture, without which efficiency of combustion cannot be a maximum.

The physical property of gases called diffusion, their activity in thoroughly intermingling with other gases to which they are introduced, is the phenomenon relied upon for the production of an economical mixture. A knowledge of the laws of diffusion of non-homogeneous hydrocarbon vapors in air, if not fully established, is essential to our problem. It is the starting point for a full solution. Given the law, every other stage of the problem is easily conformable.

A fog is probably a perfect example of the extreme of pulverization of a liquid, but the fuel is still in a liquid state. It is unquestionably an effective stage in the process of evaporation. Whether the laws of diffusion will permit the effective intermingling of the gases subsequent to the introduction of the fog to the cylinder and prior to the application of the spark, or whether the fuel must be first evaporated is to be determined. No time should be lost in the effort.

Many kinds of apparatus have been proposed. Some are already on the market and others are in course of development that aim to improve the fuel efficiency of existing and future engines. Some have been developed for the purpose of vaporizing kerosene, but in principle they are applicable, at least to some extent, to the use of gasoline, because the objects aimed at are identical, namely, the evaporation of all the fuel and the formation of a chemically active mixture prior to ignition in the cylinder.

Heating Devices Classified

Those devices may be divided into four classes:

(1) Those which heat the fuel only before mixture with the air

(2) Those which heat the air before mixing it with the fuel

(3) Those which heat all the fuel and air mixture

(4) Those which heat the fuel with some air and add cold air thereafter

Devices in the first group are, in general, subject to the disadvantage that gasoline has of being composed of fractions having various boiling points. Those fractions of low boiling point will evaporate in the carbureter before reaching the jet or exit, because they are subject to the excessive temperature necessary to boil off the high-boiling-point fractions. Thus, if the fuel was heated in a common carbureter and the float-chamber was open to the atmosphere these fractions would be lost, or, if closed, they would generate pressure and thus render abortive any attempt to meter the fuel so far as the production of the correct proportion of fuel to air in the mixture is concerned.

One form of device boils the fuel by a flame, using some of the fuel for the purpose. The resulting vapors are mixed with cold air and form a fog ready for admission to the cylinders. The many variables introduced by changes in the throttle position and in engine speed necessitate the use of a number of separate mechanisms for controlling the generation of gas in proportion to the demand. The use of fuel for evaporating the engine fuel is an economic loss, later referred to in the discussion of the third class.

Regarding the second type, the common stove or oven for heating the air as it passes over the hot surface of the exhaust pipe is a step in the right direction but, as usually fitted, it is entirely inadequate, as clearly illustrated in a recent test made by the Fifth Avenue Coach Co., on its electric dynamometer, where the original small stove could maintain only 150 deg. fahr. in the air entering the carbureter at normal full load, which was maintained by a No. 67 carbureter drill size jet. By using an oven entirely enclosing the exhaust manifold, insulating its exterior surface with asbestos and conducting the hot air to the carbureter through a well-insulated pipe, the air entering the carbureter attained a temperature of 250 deg. fahr. Black smoke from a rich mixture resulted and on substituting a No. 70 jet for the No. 67, normal full power was attained and maintained. The saving in fuel was approximately 10 per cent. Since the company referred to now uses 1,500,000 gal. of gasoline per year, this means a saving of $50,000 per year, or $5,000 for every 1 per cent. saved.

The temperature of the hot air entering the carbureter changes with atmospheric variations but more especially with the variation in the exhaust-pipe temperature and with the change of rate of flow due to throttle and engine speed variations; it sometimes fluctuates as much as 150 deg. fahr. This varies the temperature of the carbureter nozzle. A change of only 50 deg. fahr. has been shown to affect the flow of gasoline as much as 30 to 40 per cent. From this point of view, a steady temperature at the carbureter jet is one essential for economy, unless the sharp-edge or disk-type jet is proved immune to a change of fuel viscocity and is without associated disadvantages.

When the engine is accelerated it takes time to heat up the exhaust pipe to the temperature of the new condition, hence, during acceleration, the increased air to fill the demand is colder than usual. Conversely, on throttling down, the very hot exhaust pipe excessively heats the diminished quantity of air demanded by the engine. Such changes vary the amount of fuel that is vaporized and it becomes very difficult to set a carbureter so that it will not cause loading or popping back. In any case, it must be uneconomical part of the time, and in the case of city traffic this means most of the time. From this it is clear that when hot-air stoves only are used they should be arranged to furnish heat in proportion to the amount of fuel used, they should be placed as near the engine port as possible and the wall between the exhaust gases and the air should be thin, preferably having fins or pins projecting both into the streams of exhaust gas and air. The stove should present the air to a large surface of the exhaust manifold in the endeavor to attain high air-temperature under all conditions with a control to adjust the temperature of the air to the requirements of the mixture.

Devices of the third class are of the more or less familiar hot-spot or hot manifold type, in which the heat of either hot water or the exhaust gases is applied to the surfaces of the inlet manifold. Hot water is entirely inadequate, for its temperature cannot be above 200 deg. fahr. and it is already shown that this temperature can do no more than boil off about 20 per cent. of the gasoline passing through the manifold.

Hot exhaust gases are the logical source of heat for the purpose, because their range of temperature will not fall below about 450 deg. fahr. while under load they will rise to about 1500 deg. fahr. If unused, their heat is a total loss amounting to some 40 to 50 per cent. of the total heat value of the fuel. With such a source of otherwise waste heat, it seems unnecessary to burn any fuel specially for evaporating the engine fuel as is done in some forms of carbureter.

The normal variation of exhaust-gas temperature broadly follows the demand for power, but for a given volume of mixture flowing to the engine per minute, calling for a definite amount of heat for the vaporization of its fuel content the heating surface may be under either of two extreme conditions; one, in which the engine is racing under no load and delivering most of the fuel heat to the exhaust and so to the fuel heating surface, or the other, in which full torque is being delivered at low speed and a minimum proportion of heat is available for the fuel-heating surface. In the first case, high vacuum in the inlet tract aids vaporization, while this condition is absent in the second case. If the heating surface is proportioned to fit the second case, a highly attenuated mixture, skipping and backfiring, will attend the first condition, especially if the carbureter is adjusted to furnish an economical mixture under load.

As in the case of the air-heating stove, a more rapid reaction to the change of throttle is obtained when the mixture-heating surface forms a part of the exhaust manifold, the wall being thin and studded or ribbed on both sides so as to readily absorb and distribute the heat to the mixture. One investigator finds that with a direct transfer of heat through one wall, 1 sq. in. of surface suffices for each 27 cu. in. of nominal piston displacement while if the exhaust be piped from the manifold to the heater 1 sq. in. of surface will be necessary for every 18 cu. in. of displacement. The latter design involves a much greater mass of material and is comparatively sluggish in its response to a change of condition.

All the devices included in the third class should also be arranged with means for maintaining a constant temperature of the mixture and, in addition, those surfaces upon which the liquid fuel impinges should be prevented from ever attaining temperatures much above the end boiling point of the fuel or cracking of the fuel with consequent deposition of carbon will result.

The last class includes arrangements which mix some air with the fuel and subject that very rich mixture to comparatively high temperature by passing it through a tube or similar device, usually surrounded by hot exhaust gases. The liquid is partially or completely vaporized, according to the exhaust temperature applied and the time allowed. The resulting vapor and air mixture is later mixed with appropriate quantities of cold air, usually resulting in production of a fog mixture for use in the cylinder. The controlling factors of these systems are similar to those of the other three classes. In addition, the high temperature to which the comparatively small heating unit is subjected causes cracking of the fuel and the deposition of carbon therein, the small passages easily become choked, while the carbon on the heating surface acts as a heat insulator, thus varying the rate of vaporization at any given temperature of the exhaust gases. The degree of vaporization and the proportions of hot mixture and cold air being thus varied, inoperative leanness of the mixture is eventually produced. Means for controlling the temperature applied to the heating unit, are desirable.

Losses of Fuel Outside the Engine

Much evaporation takes place in handling and transporting fuel, and more occurs while in the tanks and carbureters of vehicles because of the temperature and agitation to which the fuel is subjected. The least loss from this source occurs with vehicles arranged for pressure feed, inasmuch as the only point at which the gasoline is exposed to the atmosphere or can evaporate is at the carbureter, and then only if it is of the float type, having a float-chamber vent. The warmer the gasoline in the float-chamber and the more it is shaken through the vibration of the vehicle, the more evporation will take place. This loss alone may be several per cent. The next most important loss occurs in the vacuum tanks now in common use. They are usually placed on the front of the dash and receive a great amount of heat from the engine.

The most serious loss is in the escape of gasoline vapor from gravity tanks vented to the atmosphere. These are usually found under the driver's seat, or in the cowl, and they receive much heat from the engine, which, combined with the action of waves of gasoline surging back and forth and forming a kind of surf, present ideal conditions for evaporation.

In the tests made by the Fifth Avenue Coach Co., losses of from 6 to 8 per cent occurred in the tank alone, the variation being due to a change of the atmospheric temperature. Such a loss, added to that of the carbureter, aggravated by the inclusion of casinghead gasoline in the fuel and by the extreme atmospheric temperature of summer, may readily amount to 10 per cent which would represent, in the case of that company, a direct financial loss of $50,000 per annum, and a far greater economic loss to the community.

Such liquids as find their way into the cylinder and are unevaporated, at least in some part, get past the piston rings and, by dilution, thin the oil on the cylinder and piston walls and in the crankcase, thus destroying the lubricating quality of the oil. The usual method adopted for offsetting this condition is to use oils of a much greater viscosity than is necessary for the lubrication of the engine, with the intention that the average condition of the oil when diluted to some extent shall be about right. The method is poor, but so long as we allow this condition of dilution we have no alternative. It is wrong, first, because a blend of carefully constructed lubricating oil with kerosene is not a perfect lubricant and, second, because when the oil is first put in to the engine it is too heavy and absorbs considerable power in overcoming fluid friction, whereas, after it has become dangerously diluted, rapid wear of the engine parts takes place, absorbing power uselessly and destroying the value of the engine.

The losses in connection with the dilution of lubricating oil are of general economic nature, principally in increased cost of upkeep and depreciation of the engine.

Various fuels, if vaporized and mixed with air in chemically correct proportion, have characteristic temperatures at which they will unite with the oxygen of the air and burn. This temperature appears to be little affected by the pressure at which the mixture is held.

According to Holm, the following are the temperatures for various fuels:

FUELS	DEG. CENT.	DEG. FAHR.
Coal Tar Oil	580	1,076
Benzol	520	968
Ethyl Alcohol	510	950
Gasoline	415	779
Kerosene	380	716
Gas Oil	350	662

Because the pressure of compression gives us a corresponding temperature of the compressed mixture, it is impossible without preignition to employ a compression which raises the temperature to 800 deg. fahr. when we are using kerosene as fuel, whereas if we used benzol we could safely have a temperature of 900 deg. fahr. through an increase of compression. When the above tests were made, certain standards of gasoline and kerosene obtained, but today "gasoline" means more nearly what "kerosene" meant then, consequently we must use compressions whose corresponding temperatures are low enough to avoid automatic ignition of the mixture of kerosene vapor and air, or lower than 716 deg. fahr.

Much has been said about the formation of detonating compounds from the fuel and of the knocking which results therefrom, but I submit that the formation of a true gas prior to its entry into the cylinder eliminates this difficulty, while the admission of liquid fuel in any form to the cylinder predisposes to cracking of the fuel and the production of uncontrollable detonations; in any case detonations are not a major factor in our problem.

The situation, I consider, is as follows:

(1) Eliminate the liquid fuel from the cylinder and the detonations disappear

(2) For all practical purposes, it is automatic ignition which produces engine knocking and the loss of power and flexibility which results therefrom

(3) Reduce the compression to the point where the resultant temperature of the mixture will not cause auto ignition or alternatively, if it were possible, use fuel of a higher auto-ignition temperature-point and preignition disappears

Given an engine having a compression pressure which, under heavy load, causes preignition when using modern gasoline, we then have difficulty in hill climbing and poor flexibility, necessitating that it be nursed through any change of operating conditions. Such an engine is unsatisfactory to the driver and uneconomical in all respects. Feed such an engine with fuel of a higher auto-ignition point, and such difficulties disappear. On a recent test in which the fuel used was two-thirds gasoline and one-third benzol, in an eight-cylinder car having no provision for heating the mixture except a hot-water-jacket on the intake manifold, knocking was entirely eliminated, the speed increased from 30 to 35 m. p. h., and the car easily started on high gear, from rest, on the steepest pitch of a difficult hill, on which the tests were conducted. The car showed inferior general performance with gasoline, due to knocking and consequent loss of power. The benzol mixture required no "nursing" of the car; it could be handled most crudely as to spark and throttle position. Such a mixture has been found to give a miles per gallon increase of 25 per cent over that of straight gasoline in general testing through average city traffic and on open country roads.

With the compound fuel the mixture was comparatively cold when entering the cylinders and therefore, at least as to the gasoline portion, was predisposed to cracking and detonation, yet no noises could be heard. The engine was as noiseless as an eight-cylinder poppet-valve engine could be. On reverting to straight gasoline, all the old noises, difficulties and losses returned. Having raised the temperature of auto-ignition, the coincident troubles disappeared.

The corollary is that if we must use high-boiling-point gasoline, or perhaps part kerosene, with its characteristic low temperature of auto-ignition, we must resign ourselves to the use of lower compressions to avoid preignition. Theoretically, we wish to avoid this expedient; practically, we must do it.

Having gasified our fuel with heat and correctly proportioned the mixture chemically, and therefore having predisposed it to knocking due to the temperature of compression, the compression to be used will be determined by the auto-ignition point of that mixture. The mixture characteristics are determined by the characteristics of the gasoline and hence we must use a compression low enough to permit the use of the worst fuel that may be expected to be available for some years to come.

Use of Indicators

Builders of engines should install means for examining the net performance of the mixture-making apparatus, as measured by the results produced in the temperature and pressure of combustion within the cylinder. Most arguments and analysis on this subject stop short of the final answer; they carry us through physics, chemistry and metaphysics, but if the engine does not perform as expected it is due always to something else, whereas we should seek the truth in the cylinder. In short, everyone who desires to study the subject thoroughly should learn pressure conditions within the cylinder by using some pressure-indicating apparatus such as a manograph. No high degree of satisfaction can come to the automotive manufacturer in bettering the efficiency of existing engines, including those to be produced within the next two years, without such an instrument.

All those who have made real strides in this art have done it through and with such an instrument, and they will reap a proportionate reward for their perspicacity. Without it, steps can be taken and advances made, but they are the toddles of a child compared with the accomplishments of an athlete.

Some may think that the power delivered to a dynamometer or evidenced in the hill-climbing ability of a car are the really practical measures of the advantages gained by the application of a novel device in carburetion, but no accurate data for our purpose are thus obtained, for the reason that, while the combined effort of all cylinders of a multi-cylinder engine may show an improvement over past performances, it may well be that 75 per cent of the cylinders are doing better work, while the remaining 25 per cent are even worse producers than formerly and are a drag on the combination. In teamwork no member of the team must lag and each component must be examined for individual efficiency. In our problem this can only be accomplished through instruments capable of telling a true story about each member of the team, thus eliminating all argument that poor oil or improper this or that may account for discrepancies.

For bettering the efficiency of existing engines, the logical procedure is as follows:

(1) Take gasolines from various sources having an end-point say 100 deg. fahr. higher than we use today, make chemically correct mixtures in the test bomb and experimentally determine the auto-ignition points of our future fuels

Good

(2) Adopt compressions in engines now being produced which will not result in auto ignition even when the temperature of the mixture, due to evaporating apparatus, is high

(3) Adopt evaporating apparatus to engines now in production for heating the mixture between the carbureter and the engine to such a point that the diffusion of the fuel gases and the air is complete before the spark is applied. Arrange to furnish such evaporating apparatus and special pistons or connecting-rods for compression adjustment for engines now running, where their number is high enough to warrant such provision

(4) Adopt means for heating the air going to the carbureter so that it maintains a fixed temperature in general operation

(5) Adopt means for automatic control of the temperature of the mixture

(6) Adopt carbureters giving accurate metering throughout the ranges of engine speed and throttle opening

(7) Adopt provisions in the carbureter to produce intense agitation of the fuel particles during the starting period only when the engine is cold

(8) Avoid evaporation of the low-boiling-point fuels through the vents of gravity or vacuum tanks or carbureter bowls

(9) Arrive at all determinations with the aid of a manograph, thus reading the ultimate answer directly from the pressures produced in the cylinders and avoiding the confusion of facts with appearances which always results when the basic factors are not readably before us

(10) Educate the public to avoid impatience when starting and to allow sufficient time for this operation, thus avoiding improper readjustments of carbureters and uneconomical use of fuel.

For confirmation of some of the engineering facts herein presented reference should be made to papers presented by Herbert Chase[2] and by Dr. Lucke[3]. These papers are full of the meat of the situation. They are strongly recommended as good reading. The present paper merely titrates a little pepsin for digestive purposes.

[2] See *S. A. E. Transactions*, vol. 7, part II, page 140.
[3] See *S. A. E. Transactions*, vol. 11, part II, page 118.

COUNCIL ACTION ON MEMBERSHIP MATTERS

AT the meeting of the Council held on Dec. 10, the following were present: President Charles M. Manly, First Vice-president B. B. Bachman, Second Vice-presidents E. H. Belden and J. J. Amory, Councilors David Fergusson, L. S. Keilholtz and J. V. Whitbeck and Treasurer C. B. Whittelsey.

Applications for membership and for Student Enrolment totalling 220 were approved, these being for Individual Membership 206, Affiliate Membership 5 and for Student Enrolment 9 respectively. Transfers in grade of membership were approved in the case of twenty-one applications.

The January meeting of the Council on the 5th was attended by President Charles M. Manly, First Vice-president B. B. Bachman, Second Vice-presidents J. J. Amory and E. H. Belden and Councilors David Beecroft and L. S. Keilholtz.

The Council approved ninety-two applications for Individual Membership, two for Affiliate Membership and one for Student Enrollment. Applications for transfer in grade of membership to the number of fifteen were approved, all of these being to the Member grade.

Needs in Engine Design

By Frank H. Trego[1] (*Member*)

Annual Meeting Paper

I THINK it can safely be stated that present-day engine designing has as its principal problem that of devising ways and means for utilizing the quality of the fuel now on the market and with which the engine must operate in a satisfactory manner. Practically speaking, there is no choice as to the grade of fuel employed.

Through Crankshaft Bearing Bolts

The older engine designers have all had sufficient experience to avoid mistakes in laying out the engine and its various parts. However, from the experience we have had with aeronautic engines in the last four years, I think the supporting of the crankshaft bearings by hanging them from the aluminum crankcase is a thing of the past and that we should all follow the practice of running the crankshaft bearing bolts clear through the crankcase and the cylinder base, using them also as holding-down studs for the cylinders. The bolts should be in the form of large-size studs, threaded into the case at the lower end to prevent the crankshaft from falling down when the cylinders are removed. The effect of this method of support is to use the fragile aluminum case as a spacer and to tie the bearings firmly to the cast-iron cylinder-block, which is much stronger and stiffer for the purpose. The bearings are thus held firmly in line and because there is less spring in the support they wear much longer and the original clearance is thereby maintained.

Aside from this item I wish to devote this paper to the fuel problem and so will give the results of considerable experimenting with the six-cylinder engines that we have recently brought out. The fuel difficulty lies in our inability to supply a proper mixture to the cylinders at low speeds with full throttle or at moderate speeds and part throttle. I refer to engine speeds corresponding to car speeds of from 5 to 15 m.p.h. Condensation takes place in the intake passages on the way to the cylinders; the gas failing to arrive in proper condition and quantity to give even running on all cylinders in a multiple-cylinder engine, at least where there are more than four cylinders. The problem consists of carrying the mixture in a proper gaseous state from the carbureter into the cylinder without having the fuel deposited out on the way. To maintain the mixture in a gaseous state, some carbureter manufacturers call for hot air delivered to the carbureter intake; but if this is accomplished at a temperature sufficient to prevent condensation beyond the carbureter, the loss in power is prohibitive because of the decrease in volumetric efficiency. We find that at low engine speeds the condensation seems to be aggravated to some extent, under certain conditions of road travel. For example, take a three-passenger car, weighing 3300 lb. and having a gear ratio of 4 to 1 and 33-in. wheels, equipped with our own engine, which is an L-head type with a 3½-in. bore, 5-in. stroke and a 1¼-in. carbureter. Such a car will, when starting at 20 m.p.h., go over the top of a grade rising 500 ft. in ¾-mile at a speed of 25 m.p.h. without feeding hot air into the carbureter; while by making no change other than adding hot air we are just able to get over the top at about 8 m.p.h. On maximum speed tests over a level concrete road we obtain 65 m.p.h. without hot air and 55 m.p.h. with it.

Application of Hot Air to Carbureter

It should thus be evident that taking hot air into the carbureter intake to overcome condensation is not good practice and some other means must be found to prevent condensation in the intake passages. Where it is impossible, without too much change in the design, to apply heat between the carbureter and its cylinders, a change in the valve-timing will assist greatly. In this case the intake valve on a six-cylinder engine should be closed 15 deg. after bottom center and not at 30 deg., as is the usual practice. In 1913 I found this to be of great benefit in the case of Packard engines, which "loaded" badly on hills at low speed with wide-open throttle. The change in valve-timing eliminated the condensation at that time, and did not affect the maximum speed. We were still able to run 72 m.p.h. with the model 48. The early timing is beyond the experimental stage and has become standard practice on all our six-cylinder models.

This timing is not sufficient to prevent the condensation of the grades of fuel which are now generally sold; to obtain adequate performance we are now obliged to apply heat to the mixture after it leaves the carbureter.

The following table gives the power with and without hot air, other conditions being the same. The figures are averages of three runs each on an electric dynamometer.

Engine Speed, r.p.m.	Power Developed with Hot Air, hp.	Power Developed without Hot Air, hp.
400	9.3	12.1
600	13.2	18.7
800	19.7	25.6
1,000	26.1	32.7
1,200	32.2	38.6
1,400	39.4	44.8
1,600	46.8	50.0
1,800	51.3	55.3
2,000	55.0	58.5
2,200	55.5	61.0
2,400	54.0	57.6
2,600	52.0	56.5
2,800	46.9	55.3

The low speeds would have been even worse with hot air, if they had been maintained for some time, as in the case of climbing a long grade, but the figures shown are probably the same as on the road for high speeds.

With the 15-deg. late closing of the intake valve, this six-cylinder engine performed satisfactorily on the road until cold weather came on, but we then found that the engine would jerk badly when running at 10 m.p.h. with throttle cracked and "load" to a considerable extent on a long slow pull with a full throttle.

Location of Intake Manifold

The entire intake manifold of the engine lies within the water-jacket from which insufficient heat is absorbed to prevent the missing of the cylinders farthest from the carbureter. Where the exhaust ports pass over the

[1] Vice-president and general manager, Trego Motors Corporation, New Haven, Conn.

intake manifold "hot spots" are formed; but the heat from these proved insufficient for the purpose and a final change was made whereby the exhaust manifold was disposed alongside of the intake manifold for its full length. The result was that the jerking or missing entirely disappeared and the car performance on the road was much more lively, with better acceleration, hill-climbing and low-speed work down to 1 m.p.h., with a loss of only 4 hp., due to decreased volumetric efficiency. This small sacrifice was more than compensated for by the better general performance.

Another important effect of the heat addition was noted in the good performance of the car with a cold engine and without the use of a choke on the carbureter intake or other device to enrich the gas mixture. Thus smoking and other detrimental effects of running a cold engine with a rich mixture were avoided. In completing the experiments, we found that best all-round performance was obtained by admitting just enough hot air to the carbureter to warm it slightly.

Care should by all means be taken to avoid any down grade in the gas passages after they leave the carbureter, for condensation is sure to take place with low vacuum unless a very high degree of heat is applied. This is, no doubt, the reason that in some V-engines raw gasoline goes into the crankcase past the pistons, for as soon as the throttle is opened the raw deposited gasoline runs into the cylinders instead of being drained back to the carbureter.

Careful experimenting should be carried on with each new engine to insure that the intake passages are as small as possible without the loss of too much power at the higher speeds, for if they are made of a size large enough to give maximum power at high speed they will, under average running conditions, call for the application of much heat to the mixture after it leaves the carbureter. On the six-cylinder engine above mentioned we found that a minimum diameter of $1\frac{3}{8}$ in. was required for the cored passages, which were naturally more or less rough on the inside. The compression space is 23.2 per cent of the total volume.

To sum up, the necessary features of a satisfactory engine with present-day fuel are:

(1) Closing the intake valve early

(2) Small size of intake passages

(3) Application of heat to the mixture after it leaves the carbureter

NATIONAL SCREW THREAD MATTERS

THE National Screw Thread Commission which was established by act of Congress over a year ago is scheduled to make its report on screw thread standardization in March. According to the law the report of the Commission will be submitted to Congress and to the Secretaries of the Departments of War, Navy and Commerce. Two members of the Commission, E. H. Ehrman and H. T. Herr, were nominated by this Society. Several months ago the Commission submitted its report in tentative form to the Society for comment.

The further action taken in this connection was the appointment by Charles M. Manly, then president of the Society, of a committee to collate and analyze representative and comprehensive data as to screw thread practice in the automotive industries of this country. These appointees were Paul W. Abbott, Lincoln Motor Car Co.; R. B. Smith, Packard Motor Car Co.; L. K. Snell, Willys-Overland Co.; Alexander Taub, General Motors Co., and W. K. Jamison, Domestic Engineering Co. This Special Committee on Screw Thread Standardization made a preliminary report at the meeting of the Standards Committee of the Society held on Jan. 6. President J. G. Vincent and Chairman B. B. Bachman of the Standards Committee were designated by the Council as representatives of the Society to confer with the committee.

Several conferences have been held by those most intimately acquainted with the subject among the members of the Society and a report has been formulated for transmittal to the National Screw Thread Commission, typical items treated being Classes of Fit, Basic Principles, Recommended Screw and Bolt Sizes, Thread Tolerances, Gages and Gaging Systems and Depth of Thread Engagement. Past-president Manly has taken an active part in the proceedings of the committee. The importance of the whole matter is, of course, obvious. The amount of valuable study that has been conducted on behalf of the Society is very gratifying and the members are much indebted to the members and specialists who have been giving attention to the work.

Truck and Tractor Meeting

THE Truck and Tractor Meeting of the Society held at the Hotel La Salle, Chicago, Ill., on Jan. 28, was a distinct success. The papers presented at the morning and afternoon sessions held closely the attention of the members, of whom there were more than 200 present.

The discussion was opened with a statement by A. H. Edgerton on the Use of Aluminum in Decreasing Unsprung Weight of Motor Trucks, particularly with reference to wheels. Following this Col. L. B. Moody of the Ordnance Department, presented a paper prepared by Past-president George W. Dunham on Artillery Motorization as Related to Caterpiller Traction. This paper elicited remarks by several members, the principal point of interest being the matter of the best policy for the Ordnance Department to follow in securing engines for installation in ordnance automotive equipment; that is, whether engines produced for commercial use should be used, or a line of engines of the various powers needed should be specially designed and standardized for the Government equipment in question. Prof. L. W. Chase of the University of Nebraska described in connection with his paper on Tractor Testing from the User's Standpoint the special testing plant and field which have been developed by a board of engineers under his supervision for making pulley and drawbar horsepower tests of tractors as provided for in Nebraska by law. The testing system that has been formulated includes the use of a towing dynamometer of the electric type.

During the afternoon session a valuable and comprehensive amount of information on the use of pneumatic tires on motor trucks of various capacities was given. S. V. Norton presented a paper entitled Relation of Solid and Pneumatic Tires to Motor Truck Efficiency, and C. M. McCreery treated the Design of Pneumatic-Tired Trucks. The technical sessions were concluded by the presentation and discussion of a paper by C. A. Norman and B. Stockfleth on a recent tractor engine test conducted by them. The authors each discussed features of the paper.

The dinner held in the evening, with over 600 present, was a happy occasion, music being provided by the Apollo Club. President J. G. Vincent introduced as toastmaster of the evening Fred Glover, who spoke directly to the point and amused the members greatly by relating briefly his experience in Washington in the Government service during the war. The members were honored by the presence of Major-General Leonard Wood, who gave a short, impressive address. E. J. Gittins, chairman of the tractor and threshing machinery division of the National Implement & Vehicle Association, discussed in an illuminating way the fundamental points involved in power farming.

Supplementing the remarks which had been made at the professional session in the morning by Col. C. L'H. Ruggles, chief of technical staff, Ordnance Department, Major-General C. C. Williams, chief of ordnance, stated in a very frank and straightforward way the great necessities of real military and industrial preparedness. He covered the essential points of the policy of different nations in this connection and endorsed highly the service of members of the Society and of its Committee in direct work and cooperative procedure with the Ordnance Department in connection with the development, design and production of ordnance automotive equipment.

An extended account of the sessions of this very successful meeting will be given in an early issue of THE JOURNAL.

The Measurement of Vehicle Vibrations

By BENJAMIN LIEBOWITZ[1] (*Member*)

ANNUAL MEETING PAPER *Illustrated with* PHOTOGRAPHS AND DRAWINGS

THE performance of a motor vehicle as a whole is determined by five fundamental criteria,[2] as follows:

(1) Range of operating speed
(2) Acceleration
(3) Hill-climbing ability
(4) Fuel economy
(5) Riding comfort

Methods of measurement have been thoroughly worked out for the first four of these, but the fifth has hitherto been left to shift for itself.

There are in general two classes of vibrations that affect the riding qualities of a car: (*a*) those originating from the road and (*b*) those originating in the chassis itself, such as engine vibrations. Each of these may be resolved into vertical, longitudinal and transverse components. From the standpoint of riding comfort the most important are the vertical vibrations, due to the road, and this paper deals with these.

The usual method of testing the riding qualities of a car is to ride in it and find out how it feels. But unfortunately this depends largely upon how the rider feels. Furthermore, in such tests there is always occasion for a difference of opinion. The main difficulties lie in first trying to estimate the degree of discomfort and then trying to remember it.

Engineers have long recognized the complete unreliability of the "feel" test, and some efforts have been made to work out better methods. Thus a water barrel or similar device has been used and the unit of measurement was then the amount of water left in the barrel after a test run. But large slow swings of the car body can obviously have more effect in this method than a series of short sharp shocks that would be more disagreeable to the passenger.

Attempts have also been made to use a spring-supported weight, but as will presently appear these instruments fell far short of meeting the requirements and gave a resultant curve in which the vehicle-body vibrations and those of the weight were hopelessly intermingled.

Photographic methods have been used both in the laboratory and on the road, but they are very laborious, slow and expensive; furthermore, owing to the thickness of the line on the plate and the necessarily large photographic reduction, only the large motions of the body can be successfully examined.

No determined effort appears to have been made heretofore to produce an instrument that could be set in the car and driven over an ordinary road, and which would accurately record the full-size vertical motions of the vehicle body. To meet this situation, particularly in connection with my own work on spring suspensions, I have developed a seismograph which traces the time-displacement curve of a point in the vehicle body. From this curve the vertical velocities, accelerations, amplitudes, periods and damping, in short, all the most important physical factors that determine riding comfort, can be accurately measured.

UNDERLYING PRINCIPLES OF SEISMOGRAPH DESIGN

The problem of designing a seismograph for vehicle vibrations is fundamentally the same as that for earth movements, namely, to provide a point that shall remain substantially at rest while things about it move. From a practical standpoint, however, the two problems are quite different. To clarify the principles of the vehicle seismograph, the means used by seismologists for recording vertical vibrations may be briefly described. A vertical-component seismograph consists simply of a mass, or "inertia element," hung from a long helical spring and connected by amplifying levers to a marker,

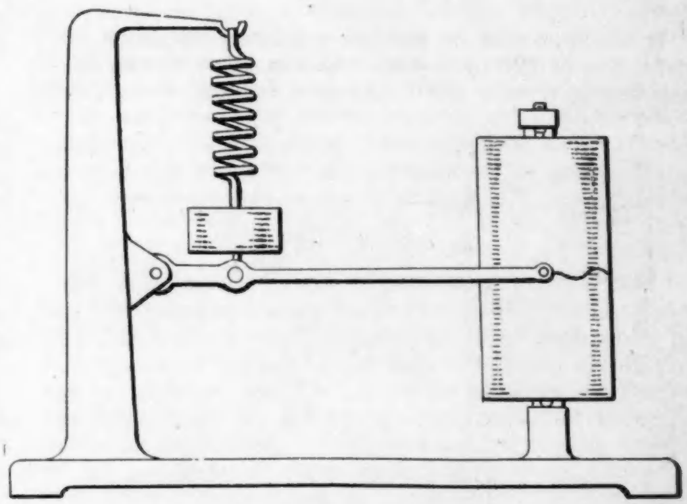


FIG. 1

which traces the relative motion between the weight and the earth on a slowly rotating smoked drum. The mass is allowed to move as it will in a horizontal plane; but since the motions are small, there is no difficulty in making the connections so that only the vertical vibrations affect the marker. (See Fig. 1)

The basic theory is this: If the suspended weight has a sufficiently long period of vertical oscillation, the frame of the instrument, including the point of support of the weight, will partake of the vertical vibration which the surroundings are undergoing without imparting any appreciable movement to the weight itself. In other words the weight or inertia element provides a "dead-point," relative to which the vibrations of the instrument, and hence those of the surroundings, can be recorded. Of course, the inertia element cannot be maintained absolutely at rest under the disturbances due to the motion of its point of support. The question therefore arises: How long must the period of the inertia element be in order that its motions shall be negligibly small?

The mathematical investigation of this problem is given in Appendix I. The net results may be stated thus:

[1] Chief engineer, Monidyne Vehicle Suspension Co., New York City.

[2] See paper on Methods of Comparing Automobile Performance by Walter T. Fishleigh in Part I, 1917, *S. A. E. Transactions*, page 259.

The period of the inertia element must be long in comparison with the longest period of oscillation that it is desired to record. In the appendix it is shown that if the period of the weight is five times the longest period of oscillation to be recorded, the error introduced is 1 in 25 or 4 per cent. This is entirely satisfactory for all practical purposes. In the appendix it is also shown that the inertia element will undergo a free oscillation whose amplitude is one-fifth of the amplitude of oscillation of the vehicle body, if the ratio of periods is 1 to 5, as before. This free oscillation is of no inconvenience whatever, as shown in the appendix, because it amounts to nothing more than a slowly-wandering zero.

The longest free period we meet with in automobiles can reach 0.7 sec. or more; hence, from the foregoing considerations, a seismograph for motor vehicles should have a free period of 3.5 sec. or longer. The static deflection which corresponds to a 3.5-sec. period is computed from the well-known period formula:

$$\text{Period} = 2\pi\sqrt{\frac{M}{K}} = 2\pi\sqrt{\frac{\delta}{g}}\text{ , where}$$

M is the mass of the inertia element
K is the stiffness of the spring, and
δ is the static deflection

It develops that to provide a 3.5-sec. period, a static deflection of 120 in. is required. In other words, if this seismograph were built like the designs of the earth

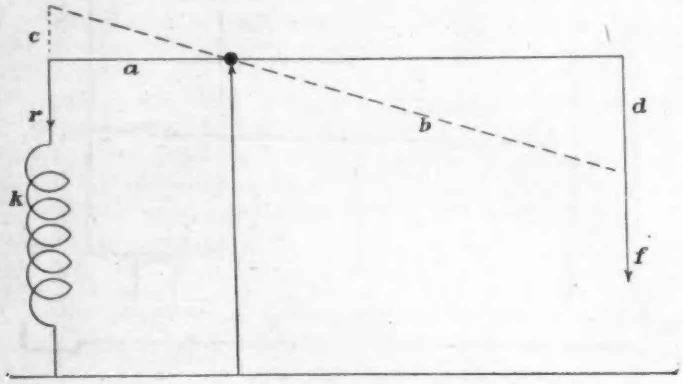


FIG. 2

instruments (Fig. 1), we would have a structure about 14 ft. high. The practical problem is therefore to provide a large deflection within a small space.

A simple lever offers a satisfactory solution. The lever suspension consists of a pivoted rod (Fig. 2) whose arms are a and b inches long. A spring is fastened at one end and a load f is applied at the other. In Appendix II the following two theorems are derived:

(1) The lever suspension acts like a simple spring whose stiffness is equal to the stiffness of the actual spring divided by the square of the lever-arm ratio

(2) The lever suspension acts like a simple spring whose static deflection is equal to the static deflection of the actual spring multiplied by the lever-arm ratio

In other words, the lever suspension multiplies the static deflection of its spring by the lever-arm ratio.

To use a practical example, suppose the lever-arm ratio is 20 to 1 and suppose that, when the lever is horizontal, the coil spring is stretched 6 in.; then the lever suspension will act like a simple spring whose static deflection is 6 times 20 = 120 in.

The lever suspension is a very old device, but its principal advantage of furnishing a large deflection in a small space does not appear to have been fully realized, and, as far as I know, this feature has not received recognition in seismograph design. This form of suspension was tried out some time ago as a vehicle spring, but although it offers interesting design possibilities, its main advantage cannot be utilized in this application. For the static deflection is limited, not by what the spring-makers can furnish, but by the clearance which the chassis-designer provides. We cannot use a 7-in. deflection with a 3.5-in. clearance and have a car that will perform satisfactorily on rough roads, unless the spring has a rising characteristic, that is increasing stiffness with increasing deflection.

GENERAL DESIGN CONSIDERATIONS

Having found what period is necessary for a practical vehicle seismograph and how to obtain that period in a practical manner, the remaining design factors to be considered are the range of movement, the weight of the inertia element, friction and the manner of suspension.

When the going is fairly rough, the movement of an automobile body at a point over the rear axle, either up or down from the neutral position, will frequently reach 3 in. or more. To care for such movements, plus the slightly-wandering zero, a range of movement of about 10 in. should be provided in the seismograph. A larger range would seldom be made use of and would add considerably to the weight and bulk of the instrument.

The weight of the inertia element is determined by frictional considerations. Suppose the friction was constant and that its total value, measured at the inertia element, was 1 per cent of the weight of the inertia element. Then the acceleration necessary to overcome this friction would be 1 per cent of the acceleration due to gravity, 0.32 ft. per sec. per sec. The instrument would then have a constant error in acceleration of 0.32 ft. per sec. per sec. This is negligible, for it requires an acceleration of several feet per second per second to be at all perceptible.

The friction in the pivots increases with the weight of the inertia element but can be minimized by the use of knife-edges. The pencil or marker friction, however, is a very variable factor. To make sure that it shall always be small compared with the weight, the latter should be large, say 10 lb. This does not materially increase the bulk of the instrument, which is determined principally by the period, the range and the requirements of rigidity.

But it should be pointed out, in passing, that sufficient friction for damping will exist in the pivots, rollers and marker. Moreover, the last can be varied so that some control of the damping is readily possible.

The manner of suspending the inertia element remains to be considered, as to whether it should be guided so as to move in a strictly rectilinear path, or rigidly fastened to the long arm of the lever suspension so that its path will be the arc of a circle. In this connection we must consider the "kick" due to the inertia reactions of the rotating parts inherent in the lever suspension. It is unnecessary to investigate this matter here, except to point out that if the inertia element is guided so that it moves up and down without rotating, and the supporting

levers are made very light, any errors due to rotary inertia reactions will be negligible.

Detailed Description of the Instrument

Fig. 3 is a photograph of an earlier form of the seismograph; Fig. 4 is the present form. In the light of the foregoing discussion, only a brief description is necessary. A is the inertia element, equipped with ball-bearing rollers. Fastened to the rigid post C is the hardened and ground guide B provided with grooves in which the rollers on A operate. The inertia element is connected to the long arm of the rocker-pivoted aluminum lever F by the connecting-rod E. The spring H and the adjustment G complete the lever suspension. The rolls J and L and the electric motor K are the principal parts of the paper mechanism. N is the marker attached to the inertia element A. A hard pencil lead is fairly satisfactory for the purpose, but a brass stylus and prepared paper are better. Ink can be used, but it does not give a fine line and is troublesome.

The paper speed is adjustable up to 9 or 10 in. per sec. High paper-speed is desirable, because the more the record is drawn out the more accurately it can be analyzed. A half-second clock and a time-marker, electrically operated by a connection with the clock mechanism, indicates half-seconds directly on the paper. This makes it possible to determine accurately the speed of the paper and to take into account variations in the speed.

The mode of operation of this instrument can be illustrated by an analogy. Imagine a test road with an accurately-built fence running alongside and a marker rigidly fastened to the body of the vehicle under test. Let the automobile be driven along the road at a constant distance from the fence, so that the marker always touches it. With the fence unaffected by the road vibrations due to the passage of the vehicle and with a constant speed, a fairly accurate record of the vertical vibrations of the vehicle body will be obtained. In the seismograph, conditions are in effect inverted, as if the marker was attached to the fence and the record-strip or paper-roll was fastened to the car body. The record traced by the seismograph is therefore inverted. When the car body moves up the paper-roll likewise moves up and, since the weight with marker attached is stationary, the slope of the line traced is downward.

The record traced by the seismograph shows the up-and-down oscillation and vibration of the vehicle body at a glance. To realize the full value of the instrument, however, the records must be analyzed carefully. The principal thing to seek, as will be pointed out, is acceleration, but amplitude and damping of the free oscillations are also very important.

Methods of Analyzing Curves

The seismograms can be roughly analyzed for acceleration with ordinary drafting instruments. Thus, in Fig. 5 let the curve represent an enlarged portion of the record. Select two points p and q on the curve, where the curvature is most rapid; draw the tangents pb and qd of any desired length and complete the right-angle triangles as shown. From the time marks note the time corresponding to the lengths pa and qc. Thus, if the paper travel is 5 in. per sec. and the length pa is 1 in., then pa corresponds to 1/5 sec. Divide the distance ba, in feet, by the time pa in seconds. This gives the vertical velocity of the vehicle body at the point p, in feet per second. In the same way determine the vertical velocity at q. Call these v_p and v_q respectively. Measure the horizontal distance from p to q and, from the time marks,

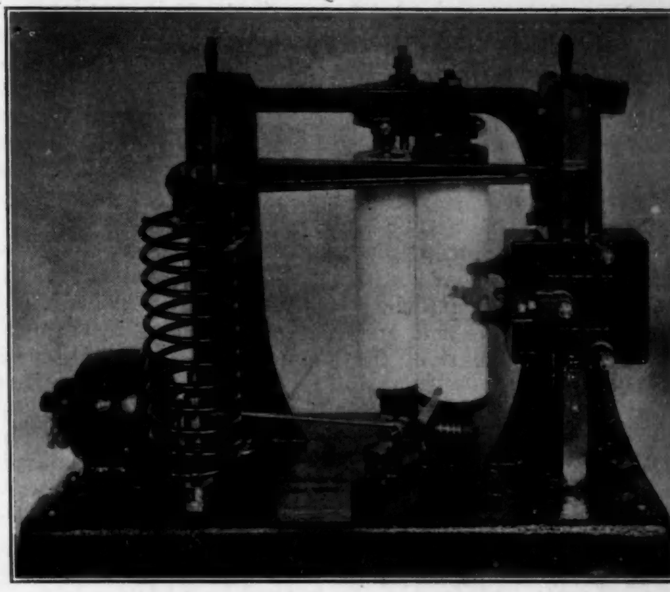


Fig. 3

determine the time t required for the paper to travel this distance. If a time-marker is not used, the time intervals can be determined from the speed of the drum, which must then be known. In the time t, the velocity of the body has changed from v_p to $-v_q$ It should be noted that the tangents have opposite slopes in the example chosen, hence the acceleration is given by

$$\frac{v_p - (-v_q)}{t} = \frac{v_p + v_q}{t}$$

By choosing the distances pa and qc, each equal to $\frac{1}{2}$ sec. for instance, the work can be considerably shortened. But this method, while satisfactory for rough work, has disadvantages; (a) it is not very accurate; (b) it gives the average and not the maximum acceleration between the two points; (c) points of maximum acceleration may easily be overlooked; (d) it does not permit the drawing of the acceleration curve, except with much labor.

Mathematically speaking, acceleration is the second derivative of the curve. This can be obtained quickly and accurately by an instrument called a comparator,

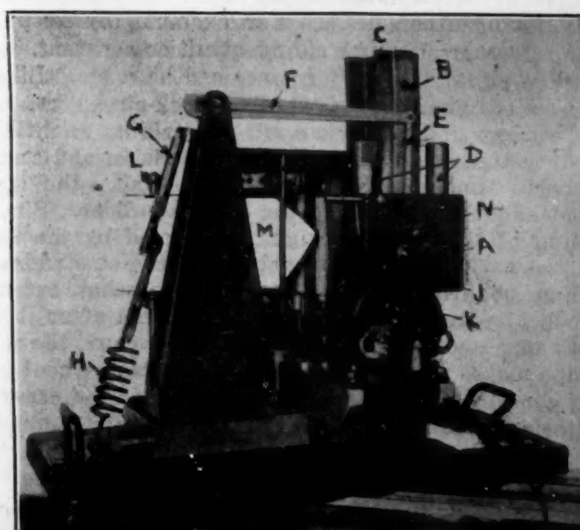


Fig. 4

such as astronomers use in measuring star photographs.

A simple form of this device has been designed specially for analyzing these curves. (See Fig. 6) It consists of a base on which is mounted a micrometer slide with its axis horizontal and parallel to a finished surface on the base. Carried by this slide is a second micrometer slide, with its axis horizontal and perpendicular to the first slide. A magnifier equipped with cross-hairs is mounted on the second slide.

In use, the part of the record to be analyzed is tacked to a drawing-board with its edges parallel to the T-square. The particular portion to be considered is brought into the field of the magnifier of the instrument, which is set square by the T-square, and is fastened with pins so that it will not shift. The cross-hairs are then set on the line near the beginning of the "hump," as at a in Fig. 7, and the micrometer readings noted. The longitudinal slide is then shifted a known small distance, a to a', depending upon the curvature; the cross-hairs are then brought back on the line by moving the transverse slide through a' to b and the micrometer readings taken. In the same way set off b to b', determine b' to c, etc. To simplify the work, make a to $a' = b$ to $b' = c$ to c', etc., $= t$, where t is the time interval corresponding to these small distances. Denote the point midway between a and b by A, between b and c by B, etc. Then

$$\text{The velocity at } A = \frac{a'\,b}{a\,a'} = \frac{a'\,b}{t}$$

$$\text{The velocity at } B = \frac{b'\,c}{t}$$

$$\text{The velocity at } C = \frac{c'\,d}{t}$$

$$\text{The velocity at } D = \frac{d'\,e}{t}, \text{ etc.}$$

$$\text{The acceleration at } b = \frac{b'\,c - a'\,b}{t^2}$$

$$\text{The acceleration at } c = \frac{c'\,d - b'c}{t^2}, \text{ etc.}$$

By properly tabulating the work, the procedure is fairly rapid. By plotting the resulting accelerations, the acceleration curve can be drawn.

While the seismograph was primarily designed for use in spring-development work, it is equally useful in a number of other fields. For example, by keeping all other factors affecting riding qualities constant, a thorough study of tire performance can be made. Likewise, tire-substitutes, spring wheels, shock-absorbers, auxiliary springs and all devices affecting riding qualities, can be investigated with a degree of precision and certainty hitherto unobtainable. But above all, the instrument should be useful to the chassis-builder. Since the riding of the car he designs is affected by the sum of all the variables, it will enable him to set standards of riding qualities and to determine to what extent his product approaches these standards, even after it leaves his hands and has been in service. In short, the seismograph makes possible a standardized measurement of riding comfort, just as the measurement of the other four criteria of automobile performance has been standardized.

Factors Determining Riding Comfort

Having found out how to measure the most important factors that determine riding comfort, the next step is to determine the relative importance of each.

These factors may be listed as follows:

(1) Vertical acceleration, both with forced vibration or a short period and free vibration or a long period
(2) Periods, both forced and free
(3) Amplitudes, both forced and free
(4) Damping, free only

There should also be included rotary acceleration, although this is not directly measurable by the seismograph, as will be pointed out.

Chassis and spring engineers have long recognized the great importance of upward acceleration. Thus, if a point in an automobile body is accelerated upward at the rate of 16 ft. per sec. per sec., or one-half the acceleration due to gravity, the passenger at that point experiences an upward force equal to half his weight; if the acceleration is 32 ft. per sec. per sec., the upward force on the passenger will equal his weight; etc. This neglects the cushioning effect of the upholstery. The stresses in the frame and in all the supporting members are likewise increased in the same proportion. There can, therefore, be no question of the basic importance of vertical acceleration, both forced and free, in the matter of riding comfort and the life of the chassis.

The question has frequently been asked, "Is acceleration the sole determining factor?" It is well understood that, speaking in terms of simple harmonic motion, the maximum acceleration is directly proportional to the amplitude and inversely proportional to the square of the period. Thus, a period of 1 sec. and an amplitude of 4 in. gives a maximum acceleration of 13.1 ft. per sec. per sec., which is about two-fifths the acceleration due to gravity. The same acceleration will be given by any one of the combinations in the accompanying table.

Period, sec.	Amplitude, in.
0.500	1.000
0.250	0.250
0.125	0.063
2.000	16.000

The oscillation in the first line of the table will give the passenger a push equal to two-fifths of his weight twice in each second; that in the second line will give him the same push four times in each second. If one faces a sparring partner, does it not matter how often one is hit as well as how hard? It follows that the first vibration will be less disagreeable than the second, even though the accelerations are the same. Furthermore, an oscillation of a 2-sec. period and a 16-in. amplitude would cause a different type of discomfort. But such large slow oscillations never occur in automobiles; they belong in the sphere of naval architecture.

Evidently the physiological effect of acceleration cannot be separated from frequency. No quantitative relations can be given at present, but in analyzing seismograms the fact should be borne in mind that, in general, the higher the frequency with which a given acceleration is repeated the harder the riding will seem.

It is sometimes argued that, since the rate of change of acceleration is of great importance in fore-and-aft acceleration, as in the starting and stopping of trains and street cars, the same should be true in up-and-down acceleration. The two questions are wholly unrelated because in starting and stopping street cars, for example, the accelerating and decelerating forces are applied to the passenger below his center of gravity and a rotating or pitching tendency is introduced; the passenger must then adjust his muscles to counterbalance this. Such acceleration must therefore be gradual enough

to allow the passenger time for this adjustment, but obviously this does not apply to vertical accelerations.

Although the rate of change of acceleration may be dismissed from a physiological standpoint, it is very important from a physical standpoint. It is not within the scope of this paper to go into this matter further. It need only be pointed out that rate of change of acceleration is directly connected with frame-whipping and body-tremors, which in turn produce local accelerations that affect the passenger.

Turning now to the question of amplitude, it is well understood that the vibrations of an automobile body as a whole are composed of forced vibrations having the period of the road shocks, and free oscillations having the natural period of the suspended body itself. The amplitude of the forced vibrations is in itself of little importance; it is the acceleration that counts.

But amplitude is of more importance in the free oscillation, because a large amplitude affects not only the sensation of stability, but stability itself. Thus, when the rear end of the body is on the high point of a large swing, the pressure between the tires and the road is

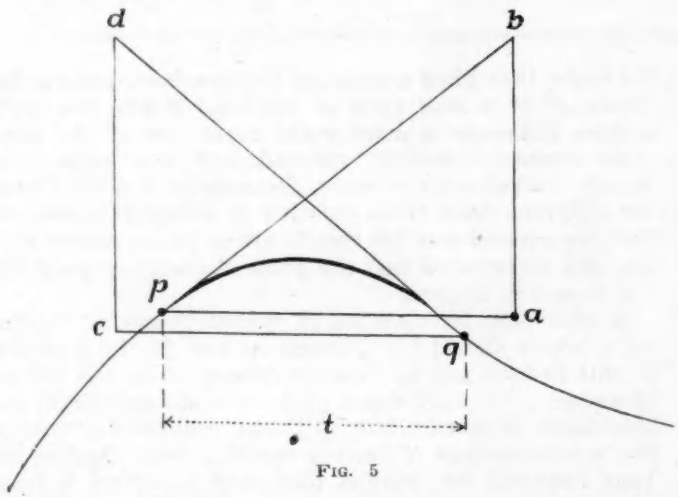


FIG. 5

diminished, with consequent decrease in the directive and tractive forces. Moreover, since the oscillations of an automobile body consist of vertical motions of the center of gravity and rotary motions about it, vertical amplitude is a measure of rotary amplitude and hence of rotary acceleration. In short-wheelbase cars these rotary accelerations sometimes administer disagreeable shoves from behind, near the shoulder-blades. This point should be kept in mind when interpreting seismograms.

While mentioning the subject of wheelbase, it might be well to point out the fallacy of certain notions which have gained currency, such as that the body periods can be lengthened by turning cantilever springs inside out, so to speak, or by sloping cross-springs away from the body center and thereby lengthening the distance between front and rear-spring anchorages on the frame. This matter can be settled by recourse to the simplest basic principles that for every action there is an equal and opposite reaction, as stated in the third law of motion. This means that the line of action of a force remains unchanged if the shape of the mechanical connections be altered. Thus, every designer knows that the line of action of the force between the wrist-pin and crank-pin is a straight line joining their centers; and no matter what shape or bends the connecting-rod may have, the line of action of the force is unchanged. Applying this principle to springs, we see that if the path of the wheel,

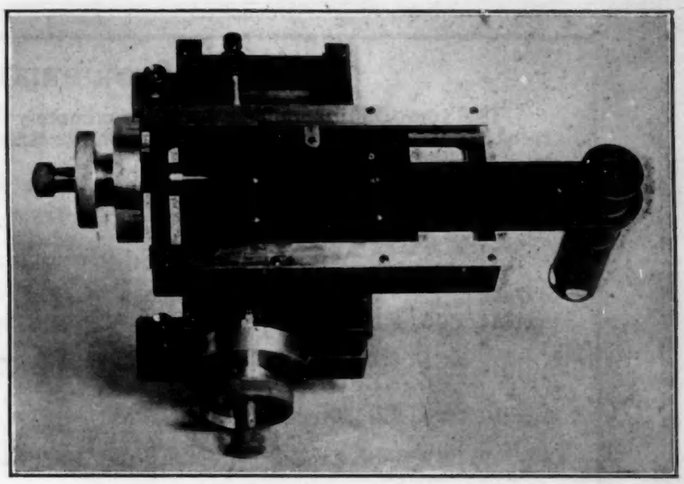


FIG. 6

with reference to the body, as it moves toward the body, is substantially a straight line perpendicular to the axis of the body, as is always the case, then the line of action of the force exerted by the spring on the wheel, which equals the reaction on the body, is substantially a perpendicular line through the wheel center. This is true no matter what form the spring takes. Likewise, the magnitude of this spring force is the product of its rate and its deflection, irrespective of the shape of the spring. It follows that the body periods are completely determined by the spring rates, the mass and radius of gyration of the body and the actual wheelbase, and it does not matter what shape the springs themselves may take.

It does not necessarily follow that such peculiar dispositions of springs will not produce better riding qualities. Thus they may permit the use of more spring material, resulting in a more flexible spring. The binding actions at the shackles, on the other hand, may result in an additional shock-absorber effect, which would have an appreciable influence on the riding qualities and an aqually appreciable influence on the life of the shackle, but the shape of the spring itself has no effect.

Turning to damping, which is directly related to friction, its function is well understood. It limits the amplitude of the free oscillations, whether due to a single impact or to accidental resonance between road bumps and body periods. In general, the greater the damping the smaller will be the amplitude of the free oscillations, however caused, and the more rapidly will they die out.

However intrduced, friction always increases the forced oscillations. The seismograph enables us to determine experimentally how much friction to add to the springs to secure the desired damping and what increase in the forced vibrations this added friction causes.

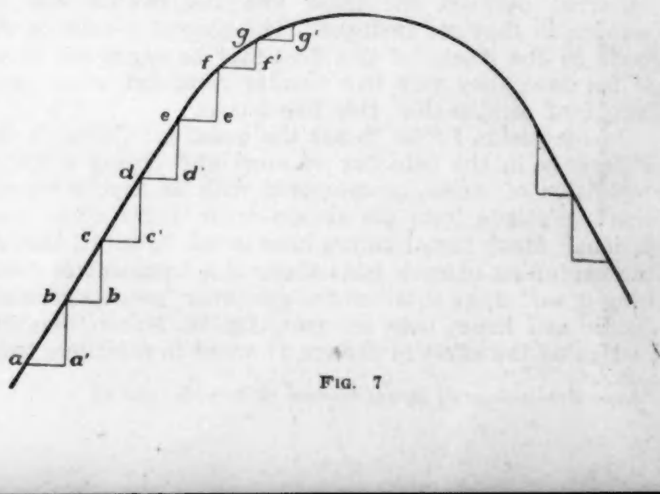


FIG. 7

DATA FOR SEISMOGRAPH RECORDS

The reproductions which are approximately half size show only a portion of each record. These records were obtained on a bus running on the streets of New York City. The period of the seismograph inertia element was 1.62 sec.

The accompanying table gives the data for the various records, the arrangement of the lines in the table corresponding to that of the curves.

Date	Street	From	To	Car Speed, M.P.H.	Paper Speed, In. per Sec.	Road and Condition	Remarks
Aug. 14, 1919	Fifth Avenue	104th Street	105th Street	15	6.3	Asphalt Medium poor	Maiden trip. Bus 506
Aug. 14, 1919	Fifth Avenue	100th Street	96th Street	12 to 14	6.3	Asphalt Medium poor	Maiden trip Bus 506
Aug. 19, 1919	Fifth Avenue	100th Street	97th Street	15	6.3	Asphalt Medium poor	Bus 506 Fifth day in service
Aug. 19, 1919	Fifth Avenue	105th Street	108th Street	15	6.3	Asphalt Medium poor	Bus 506 Fifth day in service
Aug. 26, 1919	Fifth Avenue	98th Street	102nd Street	15	6.3	Asphalt Medium poor	Bus 506 Twelfth day in service
Aug. 26, 1919	Riverside Drive	110th Street	109th Street	15	6.3	Asphalt Good	Bus 506 Twelfth day in service

Damping can be conveniently determined by driving the car over a larger bump, at a desired speed, and noting from the seismogram the number of swings the body performs in coming approximately to rest. The car can also be driven at a critical speed over a certain road and the amplitude of the free oscillations determined from the seismograms.

Summarizing, the most important factors that determine riding qualities are: The vertical accelerations, both forced and free; the frequency with which they are repeated; the amplitude of the free oscillations; the damping of the free oscillations; and rotary acceleration. All of these, except the last, can be directly determined by the seismograph.

Preliminary Experimental Results

I hope to give later a detailed account of the experimental investigation of riding qualities now under way, but I can say now that the results have definitely shown the riding qualities of a given car to vary to an extent not realized by most engineers. Thus, measurements on several trucks of a large fleet operating in New York City showed that, on a certain stretch of upper Fifth Avenue a maximum vertical acceleration of approximately one-half that due to gravity was developed on new trucks; but on trucks that had been a year or more in service the corresponding acceleration was about one and one-half times that due to gravity, a difference of some 300 per cent. A later test showed that most of this change occurred between the third and the twelfth day of service, in that one instance. The general results of the tests on the trucks of this fleet may be expressed thus: When new, they ride like touring cars, but after some length of service they ride like trucks.

In conclusion I wish to ask the question: What is the difference in the behavior of steel undergoing a cyclic repetition of stress, as compared with an acyclic repetition? Fatigue tests are always made under cyclic conditions. Many investigators have noted, however, that at the beginning of such tests there is a transient or "settling-down" stage in which the specimen "gets its stride." So far as I know, only one man, Dr. W. Mason,[3] has investigated the effect of change of speed in a fatigue test. He found that when a fatigued test-specimen had become "trained" to a slow cycle of torsional stress, the cyclic strains following a more rapid repetition of the same stress were temporarily reduced, and vice versa. Dr. Mason worked with stresses, frequencies and conditions far different from those existing in automobile springs, but in a general way his results are in entire accord with my own observation that the greatest enemy of good riding is spring-fatigue.

A wide field for research is opened by acyclic fatigue tests, which should be of great interest to the members of this Society and to all those interested in the testing of metals. Not only would such tests add greatly to our knowledge of metals, but, if I may venture a prophecy, the acyclic method of testing would greatly shorten the time required for fatigue tests and so attain a long-sought goal.

[3] See *Proceedings* of Royal Society of London, vol. 92.

Appendix I

Suppose a weight, of mass M (See Fig. 8), is hung by a spring of stiffness K, from a point of support that is undergoing a simple harmonic motion of amplitude E and period T_1. Let x be the displacement at any instant of the point of support from its neutral position, and let y be the corresponding instantaneous displacement of the mass. Also, let the static deflection of the mass be δ; in other words, the weight Mg stretches this spring an amount δ.

The force acting upward on the mass is the stretch of the spring times its stiffness. The spring-stretch is obviously the difference in the displacements x and y, plus the static deflection $(x-y+\delta)$. The force acting downward is the weight, Mg. Let us call upward forces positive; then by equating the algebraic sum to mass times acceleration we get

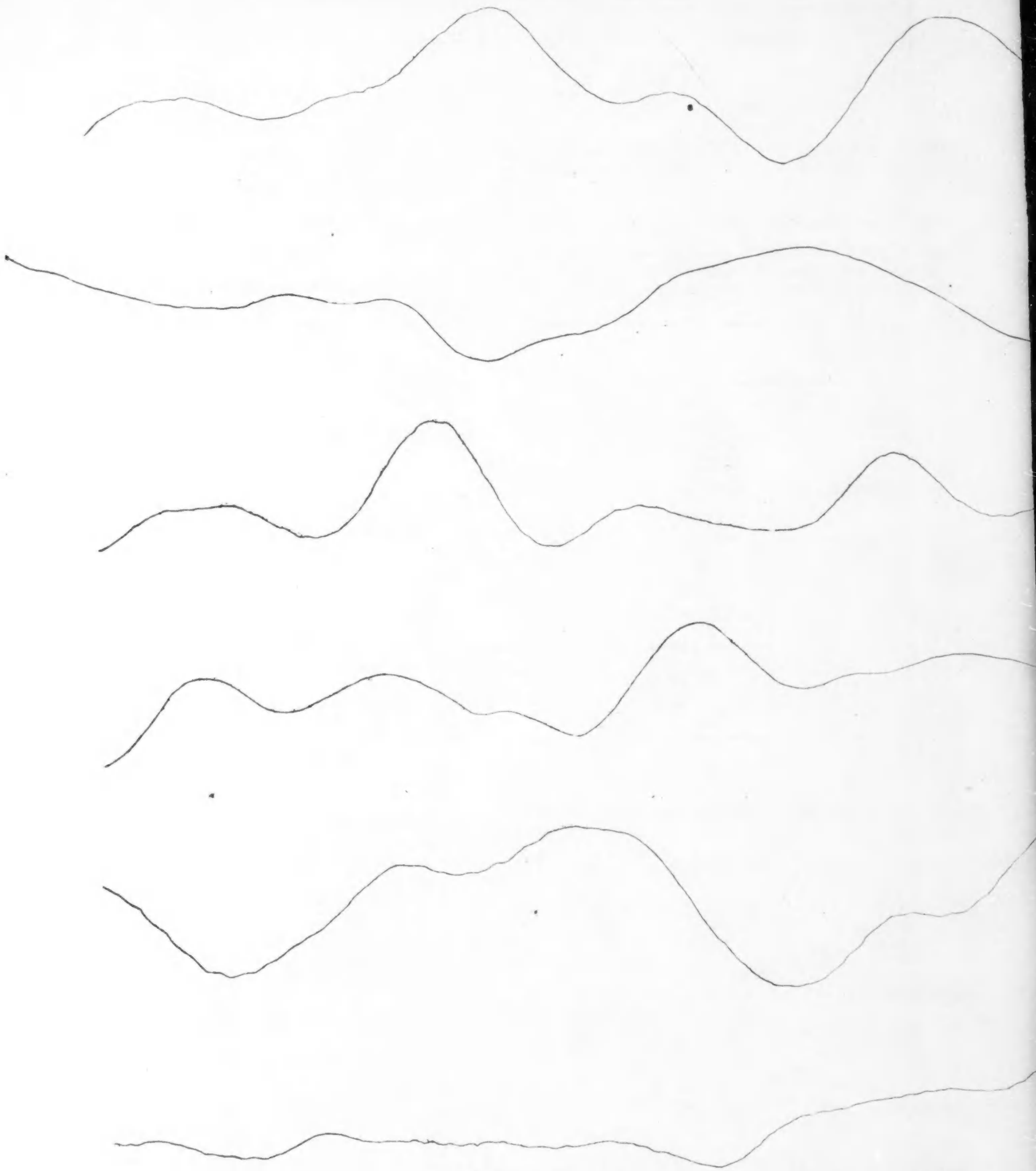
$$M\frac{d^2y}{dt^2} = -Mg + K(x - y + \delta)$$

But since δ is the spring-stretch due to the weight acting alone

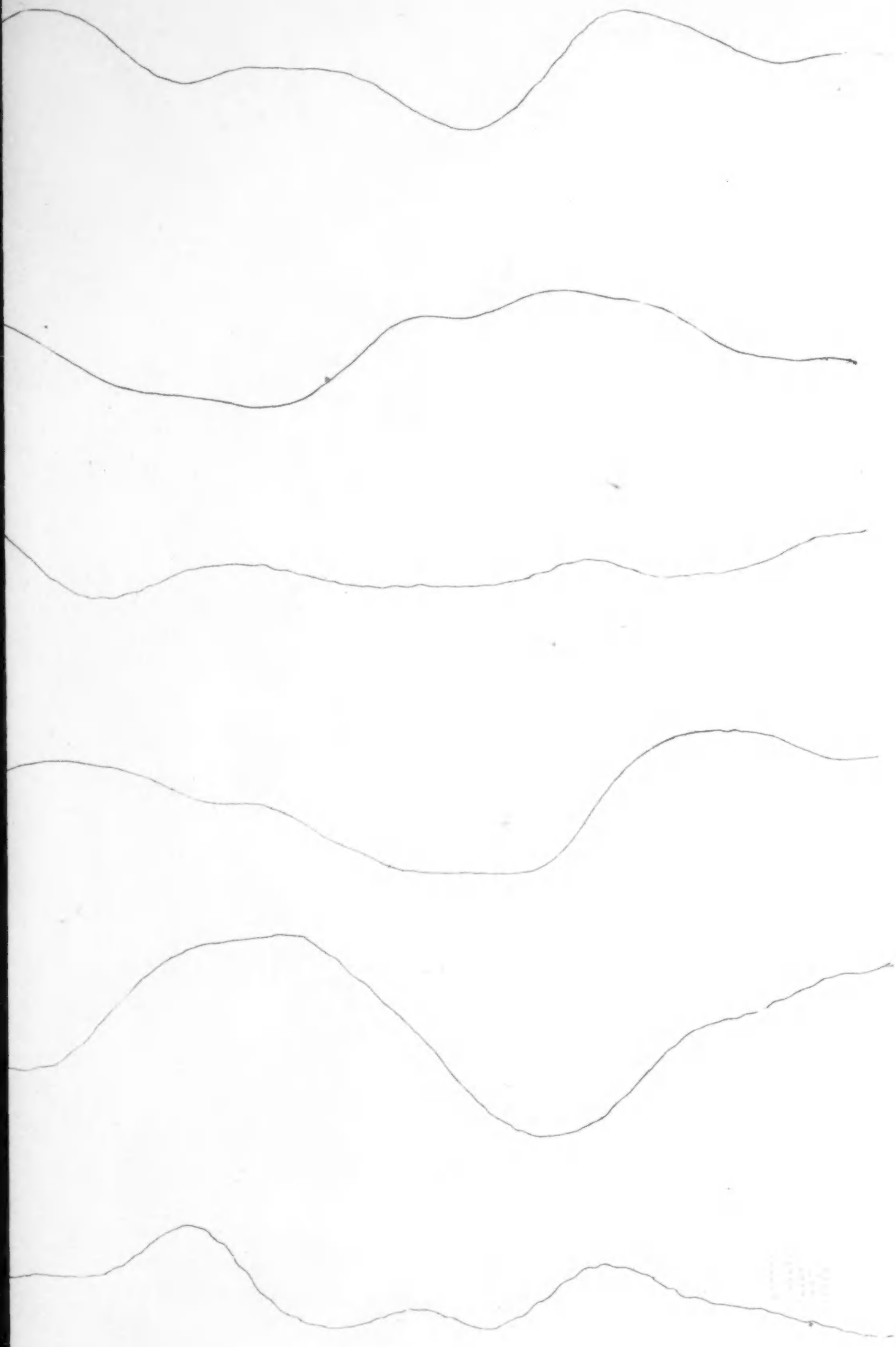
$$Mg = K\delta, \text{ hence}$$

$$M\frac{d^2y}{dt^2} = Kx - Ky, \text{ or}$$

$$M\frac{d^2y}{dt^2} + Ky = Kx \qquad (1)$$



TYPICAL CURVES OF VEHICLE BODY VIBRATION TRACED B



TRACED BY THE SEISMOGRAPH.

The simple harmonic motion of the point of support is expressed by

$$x = E \sin\left(\frac{2\pi}{T_1}\right)t \qquad (2)$$

For abbreviation, put

$$\frac{2\pi}{T_1} = p \text{ or } T_1 = \frac{2\pi}{p} \qquad (3)$$

p is called the "frequency-speed" and is equal to 2π times frequency.

Substituting in equation (1), we obtain

$$M\frac{d^2y}{dt^2} + Ky = KE \sin pt \qquad (4)$$

This is a familiar differential equation whose general solution is given in all textbooks on the subject. Suffice it to say, that the general solution consists of two parts

$$Y_1 = A \sin qt + B \cos qt \text{ (Free oscillation)} \qquad (5)$$

$$Y_2 = C \sin pt + D \cos pt \text{ (Forced oscillation)} \qquad (6)$$

The constants A, B, C, D, are obtained by substitution in equation (4), and from the initial conditions, that the weight is at rest when the motions start. Mathematically specified, the initial conditions are that when

$$t = 0, y = 0, \frac{dy}{dt} = 0$$

Substituting equation (6) in equation (4), we obtain

$$p^2M\,(C \sin pt + D \cos pt) + K(C \sin pt + D \cos pt) = KE \sin pt$$

This is true for all values of t; so put $t = 0$, which makes

$$\sin pt = 0, \cos pt = 1, \text{ and we get}$$

$$(-p^2M + K)\ D = 0 \therefore D = 0 \text{ and hence we get}$$

$$(-p^2M + K)\ C \sin pt = KE \sin pt, \text{ hence}$$

$$C = \frac{KE}{p^2M - K} \text{ (Amplitude of forced vibration)} \qquad (7)$$

This is the amplitude of the forced vibration. The period of the forced vibration is the same as the period of oscillation of the point of support. Hence the forced vibration period is given by the equation

$$Y_2 = \frac{KE}{p^2M - K}\sin pt \quad \text{(Forced vibration)} \qquad (8)$$

Referring now to the equation of free oscillation (5) the quantity q is related to the period T_2 of free oscillation by the following equation which is similar to equation (3):

$$q = \frac{2\pi}{T_2} \text{ or } T_2 = \frac{2\pi}{q} \qquad (9)$$

As is well known, or, as can be shown by substituting equation (5) in the left-hand side of equation (4) and equating to zero, the period of free oscillation is given by

$$T_2 = 2\pi\sqrt{\frac{M}{K}} \quad \text{or } T_2 = 2\pi\sqrt{\frac{\delta}{g}} \qquad (10)$$

To determine the constants A and B, we must remember that the whole motion of M is given by the sum of equations (5) and (6). Remembering also that determining the constants C and D makes equation (6) reduce to equation (8), we have

$$Y = Y_1 + Y_2 = \frac{KE}{K - p^2M}\sin pt + A \sin qt + B \cos qt$$

Differentiating

$$\frac{dy}{dt} = \frac{pKE}{K - p^2M}\cos pt + qA \cos qt - qB \sin qt$$

Introducing our initial conditions

$$y = 0, \frac{dy}{dt} = 0, \text{ when } t = 0$$

$$0 = 0 + 0 + B, \text{ hence } B = 0$$

$$0 = \frac{pKE}{K - p^2M} + qA, \text{ hence}$$

$$A = -\frac{p}{q}\frac{KE}{K - p^2M} \qquad (11)$$

This is the amplitude of the free oscillation. Hence, the expression in equation (5) for the free oscillation becomes

$$y = -\frac{p}{q}\frac{KE}{K - p^2M}\sin qt \quad \text{(Free oscillation)} \qquad (12)$$

Summarizing, we may say that: If the point of support undergoes a simple harmonic motion of amplitude E and period T_1, the weight will undergo two simple

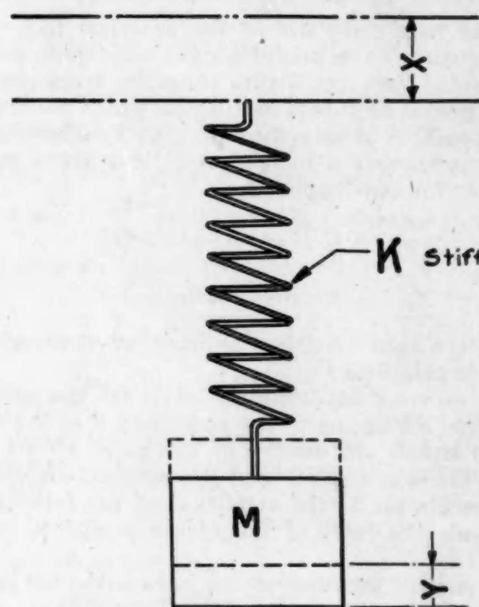


FIG. 8

harmonic motions, one of which, called the forced oscillation, will have the same period T_1, and amplitude $\frac{KE}{K - p^2M}$; the other of which, called the free oscillation, will have the period $T_2 = 2\pi\sqrt{\frac{M}{K}}$ and the amplitude $-\frac{pKE}{q\,(K - p^2M)}$

The negative sign indicates that the free and forced oscillations are opposite in phase.

The expression for the amplitudes can be simplified, as follows:

$$\frac{K}{K - p^2M} = \frac{\frac{K}{M}}{\frac{K}{M} - p^2}, \text{ but}$$

$$p = \frac{2\pi}{T_1}, \text{ or } p^2 = \frac{4\pi^2}{T_1^2}, \text{ and}$$

$$T_2 = 2\pi\sqrt{\frac{M}{K}}, \text{ hence } \frac{K}{M} = \frac{4\pi^2}{T_2^2}, \text{ hence}$$

$$\frac{K}{K - p^2M} = \frac{\frac{4\pi^2}{T_2^2}}{\frac{4\pi^2}{T_2^2} - \frac{4\pi^2}{T_1^2}} = \frac{\frac{1}{T_2^2}}{\frac{1}{T_2^2} - \frac{1}{T_1^2}}$$

Multiplying the numerator and the denominator by $T_1^2\ T_2^2$, we get

$$\frac{K}{K-p^2M}=\frac{T_1^2}{T_1^2-T_2^2}$$

In the same way

$$\frac{p}{q}=\frac{T_2}{T_1}\text{, and}$$

$$\frac{p}{q}\,\frac{K}{K-p^2M}=\frac{T_2}{T_1}\times\frac{T_1^2}{T_1^2-T_2^2}$$

Hence, our amplitudes become

$$\frac{T_1^2}{T_1^2-T_2^2}\times E \text{ (Forced oscillation), and}$$

$$\frac{T_2}{T_1}\times\frac{T_1^2}{T_1^2-T_2^2}\times E \text{ (Free oscillation).}$$

We shall now make use of the practical fact that, if such a system is to be useful for seismographic purposes, the period of free oscillation must be large compared with any period of forced oscillation which we are likely to meet, i.e., T_2 is large compared with T_1. Hence it gives a good approximation to neglect T_1^2 compared with T_2^2, and we get for our amplitudes

$$\frac{T_1^2}{T_2^2}\times E \text{ (Forced oscillation)}$$

$$\frac{T_1}{T_2}\times E \text{ (Free oscillation).}$$

The negative sign has been omitted, as it merely indicates phase relations.

Hence, we may say that the ratio of the amplitude of the *forced* oscillation to the amplitude E of the *forcing* oscillation equals the *square* of the ratio of the *forced period* to the *free period;* and the ratio of amplitude of the *free* oscillation to the amplitude of the *forcing* oscillation equals the ratio of the *forced period* to the *free period.*

In this part of the analysis we have neglected friction. We shall consider first the forced oscillation, as this is the one that causes a real instrumental error. The free oscillation, as will be shown, amounts to nothing more than a slowly-wandering zero.

From the above, it follows that, to keep down the forced oscillation and hence make the instrumental error small, the free period of the inertia element must be long when compared with the longest period of vibration of the vehicle body.

For, on account of the axle motion, the vehicle body itself undergoes forced and free vibrations and both of these are *forcing* vibrations from the standpoint of the inertia element. The longest free period of the vehicle body is the longest *forced* period on the inertia element which need be considered in practice, hence we may state as the fundamental requirement that

> The free period of the seismograph must be long compared with the free periods of the vehicle body. Or, since the periods are functions of the deflections, the static deflection of the seismograph spring must be very large compared with the static deflections of the vehicle springs.

Let us consider a practical example. Suppose a point in the vehicle body is oscillating with an amplitude of 1 in. and a period of ½ sec.; and suppose the period of the inertia element of the seismograph is 2½ sec. Then the ratio of periods is 1 to 5, and the amplitude of the forced vibration of the inertia element will be $1 \times (1/5)^2 = 1/25$ in. In other words, if the period of the seismograph is five times that of the vehicle body, the forced oscillation of the inertia element will have one-twenty-fifth of the amplitude of the free motion of the vehicle body. The greatest instrumental error would, therefore, be 4 per cent. This accuracy is entirely satisfactory for all practical purposes. We may, therefore, say that the free period of the seismograph should be about five times the longest free period of the vehicle body.

The free periods we meet with in automobiles may reach perhaps 0.7 sec. or more, hence a seismograph for motor vehicles should have a period of about 3½ sec. or longer.

Turning now to the free oscillation, we find that its amplitude is to that of the forcing oscillation inversely as the ratio of periods; so that, if the ratio of periods is 1 to 5, and the amplitude of the body-motion is 1 in., the free amplitude of the weight-motion will be 0.2 in.

To see the effect this has on the curve, suppose the zero to be stationary, and the body to be executing a free oscillation such as is shown in Fig. 9 by the light full line C. The corresponding free motion of the weight is indicated by the line W, and the resulting distortion of the record by the heavy line V.

No extended analysis is necessary to show that, when the period of the weight is long compared with that of the impressed oscillations, 5 to 1 for example, the record distortion caused by the free oscillations of the weight produces no appreciable error in acceleration, amplitude or damping. Of course, the free oscillations of the weight are damped. In the figure we have assumed them to be undamped to exhibit the worst case.

In any case, it is impossible to design an inertia-operated seismograph so that the weight will remain absolutely at rest. To emphasize this point, imagine that a sudden change occurs in the road-level (Fig. 10). As the various body-oscillations resulting from the impact die out, the inertia element, if perfectly damped, will gradually move to its new zero-position in space, which is its old zero-position with reference to the car. During the transition, the zero-position of the weight will therefore shift gradually. A wandering zero is therefore inherent in such devices. The wandering zero occurs in many types of instruments. A notable example is the syphon recorder used as a receiver in submarine cable work.

In the above analysis we have considered the vehicle body as performing one simple harmonic oscillation, whereas the actual motion is a very complicated, irregular curve. But no matter how irregular, any particular portion of the curve can be represented by a series of simple harmonic motions, called a Fourier's series. Each term of this series will give rise to a corresponding term in each of two similar series representing the forced and free motions of the seismograph weight respectively. The relations found above for a single simple harmonic oscillation will hold for each term of the series. The longest period in the series which is of importance from a practical standpoint, is the period of free oscillation of the vehicle body; and if the instrumental error is small for this term, it will be still smaller for the shorter periods.

Appendix II.

Referring to Fig. 2, imagine a pivoted rod, whose arms are a and b inches long, to have a spring fastened at one end and a load, f, applied at the other. Let k be the stiffness of the spring, and r the spring reaction or pull. When the system is in equilibrium, as indicated by the full lines

$$ra = fb.$$

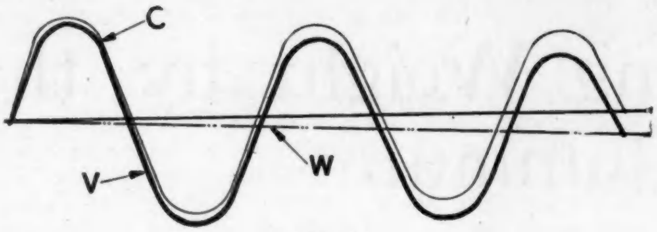


FIG. 9

Also, if the spring has been stretched a distance s to balance f, then

$$r = ks, \text{ and}$$
$$ksa = fb.$$

Let the force f be now increased a little to become $f + \Delta f$; then the system will move to a new equilibrium-position, shown by the dotted line. Note that the force f has moved through a distance d, while the spring has received an additional stretch c, the relation between c and d being obviously

$$\frac{c}{d} = \frac{a}{b} \text{ or } c = \frac{a}{b}\, d$$

The new spring reaction is r', for instance, where

$$r' = r + \Delta r = (s + c)k = r + ck$$

Taking moments about the point of support, we have

$$(r + ck)a = (f + \Delta f)b, \text{ hence}$$
$$cka = \Delta fb \text{ (For } ra = fb\text{), hence}$$
$$\Delta f = \frac{a}{b}\, ck = \left(\frac{a}{b}\right)^2 dk$$

To find the stiffness, or rate, of a spring, we divide the load by the deflection, or the increment in load by the corresponding increment of deflection. The combination of lever and spring acts as a simple spring whose stiffness is such that when the load is increased by Δf, the deflection increment is d. Hence, the stiffness of the lever suspension is

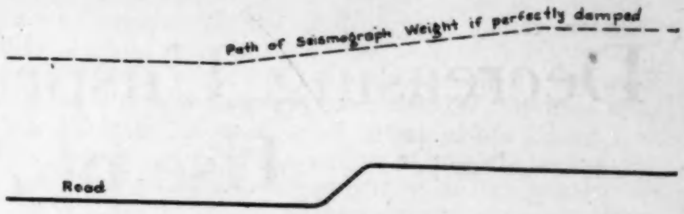


FIG. 10

$$\frac{\Delta f}{d} = \left(\frac{a}{b}\right)^2 k = k \div \left(\frac{b}{a}\right)^2$$

Hence, we obtain

Theorem I: The stiffness of the lever suspension is equal to the stiffness of its spring divided by the square of the lever-arm ratio

The static deflection of a spring is the load divided by the stiffness or $\frac{r}{k}$. If the spring of the suspension can carry a load $f\,r$, then at the long end of the lever it can carry only $\frac{a}{b}\, r$. The *static deflection* δ of the lever suspension is, therefore

$$\delta = \frac{a}{b} \cdot r \div \left(\frac{a}{b}\right)^2 k = \frac{r}{k} \cdot \frac{b}{a}$$

But $\frac{r}{k}$ is the static deflection of the coil spring k under a load r, which we denoted above by s. Hence, the static deflection of the lever suspension is given by

$$\delta = s\,\frac{b}{a}$$

We therefore obtain

Theorem II: The lever suspension multiplies the static deflection of its spring by the lever-arm ratio

PRE-WAR LIMITS IN STEEL SPECIFICATIONS RESTORED

THE automobile steel specifications of the American Society for Testing Materials have been brought into the same state of exact conformity with those of the Society of Automotive Engineers as existed prior to June, 1918. At that time the former society, to assist in meeting the unusual conditions then prevailing, adopted a war clause at its annual meeting, which was attached to and formed a part of all of its steel specifications. This clause read as follows:

"In view of the abnormal difficulty in obtaining materials in time of war, the rejection limits for sulphur in all steels and for phosphorus in acid steels shall be raised 0.01 per cent above the values given in these specifications. This shall be effective during the period of the war and until otherwise ordered by the Society."

At the annual meeting of the American Society for Testing Materials held in June, 1919, the original sulphur and phosphorus limits for automobile steels were restored as well as to the other twenty-six standard and the two tentative steel specifications of that body.

Decreasing Unsprung Weight by the Use of Aluminum

By A. H. Edgerton[1] (*Member*)

Chicago Truck and Tractor Meeting Paper

THE desirability of reducing the unsprung weight in motor vehicles is a recognized fact. Statistics recently gathered indicate that 75 of 100 engineers interviewed on this subject favor the reduction of unsprung weight, 15 expressed themselves as not knowing, while the remaining 10 per cent would not commit themselves. With so great a proportion favoring the reduction of unsprung weight we may ask what are the particular advantages to be secured? We find that the most important ones are improved riding qualities, economy in tire wear and better acceleration. No attempt will be made in this paper to present mathematical deductions which establish the most desirable ratio of sprung to unsprung weight. The intention is rather to introduce a few arguments in favor of lighter wheels and axles.

Improved Riding Qualities

An automobile body virtually floating in space and insulated from the vibratory action set up by the road irregularities would be ideal. The less the unsprung weight, the less the energy to be absorbed by the springs for a given road shock and the less the spring flexure. An ideal spring would have such resilient qualities that it would absorb and respond to the slightest shocks and at the same time possess such strength and have so wide a range of movement that it would absorb the maximum shocks. Numerous spring types have been offered as the solution of the spring-suspension problem, but with every type the fact is recognized that the effects of the unsprung weight depend primarily upon the ratio of sprung to unsprung weight.

There are, to my knowledge, no data that determine the most desirable ratio of sprung to unsprung weight. An investigation, however, of the proportional weight of the unsprung and sprung parts of cars generally considered as possessing good riding qualities show that the ratio is about 1 to 3. On a car weighing 2600 lb. and with a 1 to 3 ratio, the unsprung weight would be 650 lb. If the total weight of the car is reduced to 2000 lb. and the same ratio maintained, the unsprung weight would be 500 lb. It is safe to assume that the weight of the lightest wheel in general use, including its tire equipment, is 75 lb., or 300 lb. for the four wheels. In a car weighing 2000 lb. and having the ratio of unsprung to sprung weight mentioned, only 200 lb. of material can be used in the axles. It is impossible to make two axles weighing only 200 lb. out of any material now on the market other than aluminum for a car weighing 2600 lb. If a lighter material is used in the construction of the wheels, a more liberal weight will be allowed for the axles and their parts. By constructing both the wheels and the axles of light metal, it is within the range of possibility to maintain the ratio mentioned, assure the desired riding qualities of the car and at the same time reduce the total weight to 2000 lb.

It has been shown by mathematical calculations that it is possible to improve the riding qualities of a car by reducing the unsprung weight. The body or load of a vehicle can be suspended to advantage when the wheel is kept in contact with the ground. The higher the ratio of sprung to unsprung weight and the lighter the wheel, the better the riding qualities of the car become. To attain ideal riding qualities the springs should be designed to afford sufficient amplitude of deflection to cause the wheels to follow the contour of the road irregularities. This can only be obtained in theory by infinitely flexible springs, but in practice the flexibility of the spring motion must be restricted so that the spring may perform its functions without necessitating too much clearance between the spring and the frame. For a given suspension range the best spring is that which has the slowest periodicity, this depending upon the difference between the unloaded and loaded camber of the spring.

As we increase the speed of an empty truck, we find that road adherence becomes unfavorable and that shocks and vibrations of such empty trucks cause rapid deterioration. These shocks and vibrations frequently are responsible for the loss of time for examination and repairs to the parts. For trucks employed in rural transportation work, where heavy loads at high speeds are required, there is no better way to prevent the rapid deterioration of the engine than to safeguard it by attaining better riding qualities.

If we consider the total sprung weight of a truck as a mass which must be held in balance, we must find a value for the unsprung weight that will allow the springs to absorb the normal vibrations and hold the body in a perfect poise. If the designer establishes the weight required in the body to keep the wheels in contact with the road, the springs will tend to function as a means of keeping the wheel in constant contact with the road in the same way that a valve-spring keeps the valve follower in contact with the cam. There is no doubt that the day will come when automotive engineers will establish the best ratio of sprung to unsprung weight, and that they will then specify the weight of the wheel and the axle for any particular truck.

One of the recent interesting developments pertinent to the reduction of unsprung weight is that of the aluminum truck wheel. Both disk and spoke-type wheels have been built which are from 50 to 60 per cent lighter than wood or steel wheels. Extensive road and laboratory tests indicate that the new wheel possesses all the desirable features claimed for the wood or steel wheel and in addition embodies certain very important advantages not possessed by either of the existing types. A reduction of 50 per cent in weight was made in one case of an aluminum wheel for use on the front axle of a 3-ton truck as compared with a wood wheel and the lightest steel that could be found for the same service. Another interesting reduction in weight was found in a rear-axle housing that was redesigned for the use of aluminum and in

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which a saving of 38 per cent was attained without sacrificing the strength of the axle in any way.

Economy in Tire Wear

It has been proved that more mileage can be obtained from a light wheel than from a heavy one. This statement is supported by both laboratory and road tests. The road test covers a period of 4 years. The comparison was made on two trucks of the same model and design both operating under similar conditions. One truck was equipped with extremely light wheels; the other, with heavy wood wheels. The latter were of a conventional design and the standard equipment for the truck. During this period of 4 years three sets of tires were completely worn out on the truck with the heavy wheels, but the one with the lighter wheels is still running on its second set of tires, and judging from the last report the tires are good for a great many more miles.

The laboratory test was conducted to determine what effect a high vibrating action or constant pound would have upon the tire and the wheel. An aluminum wheel weighing 82 lb. equipped with a 36 by 6-in. tire was mounted on one end of the rear axle of a truck. On the other side of the truck a steel wheel with similar tire equipment weighing 248 lb. was mounted. The truck was then set up on a test stand which brought the wheels in contact with a large cast-iron drum 16 ft. in circumference. Cleats ¾ in. high, and spaced about 4 ft. apart, were bolted across the face of this drum. The truck was secured with chains and staybolts against forward and lateral movements and, as the drum was free to rotate, it was possible to drive the truck under its own power and at a speed to attain a very severe pound on the wheel and tire. A run of 500 miles was made at a speed of 15 m.p.h. At the conclusion of this test it was found that the tire on the steel wheel showed deep cracks across the face. These cracks were 6 to 8 in. apart and in one instance the fracture penetrated the entire depth of the tire. The tire on the aluminum wheel had small fractures around both outer edges or corners of the tire. These were about ¾ in. wide and 2 in. long, but the central portion of the tire remained as good and as solid as at the beginning of the test. The tires and wheels were subjected to 19,800 impacts per wheel per hour, or a total of 6,009,300 impacts for the whole test. It should be added here that the truck was loaded to its capacity during the test.

At the conclusion of the test a careful examination of the wheels revealed no signs of fracture or metal fatigue of any kind. An additional advantage of the aluminum wheel is the ease of tire removal where pneumatic tires are used. Aluminum will not rust and, as the aluminum wheel has a rim and side-ring of the same material, the tire will not "freeze." Everyone is familiar with the difficulties encountered in removing a tire that has been on an ordinary wheel for a long time. This operation is both tedious and troublesome, particularly when a tire change is necessary on the road. A tire engineer of one of the largest companies recently stated that this was one of the greatest difficulties the tire manufacturers had to face. While galvanizing has been offered as a solution of this problem, it is conceded that this is by no means entirely satisfactory.

Better Acceleration

In all the larger cities traffic in certain districts is so congested that a driver has to make innumerable stops, sometimes several in a single block. It is believed that in considering the acceleration of a vehicle, the point of most general interest centers in that phase of the problem which deals with the momentum of the car or truck when starting from rest.

Considerable effort has been expended in an endeavor to better the "get-away" of cars, though it appears that little attention has been given to one of the salient factors, that of reducing the weight of the wheels and the axles. By decreasing such weight we reduce proportionately the inertia that must be overcome in accelerating.

Experiments have consequently been made to determine the moments of inertia of a light and of a heavy wheel. An aluminum and a steel wheel were chosen, each being designed for a 3-ton White truck equipped with 38 by 7-in. pneumatic tires. The aluminum wheel weighed 62 lb. and the steel one 149 lb. A clamping device was made so that the wheel could be gripped at the outer edges of the rim. A bar which was secured to the clamp in a position parallel to the hub axis of the wheel supported the wheel as a pendulum and was mounted on knife-edges so that the wheel was free to swing from this suspension point.

With the wheel thus suspended we are able to find the time per oscillation in seconds and the moment of inertia can then be obtained by the formula

$$I = \frac{mrgt^2}{\pi^2} = \frac{wrgt^2}{g\pi^2}, \text{ where}$$

I = the moment of inertia in inch units

m = the mass or $\frac{w}{g}$

r = the radius of the wheel in inches

g = the force of gravity or 32.2×12

t = the time in seconds for a single oscillation

It was found that the moment of inertia of the aluminum wheel was 53 and that of the steel wheel 123.

For any power-driven vehicle the force needed to produce a given angular velocity of the wheels in a given length of time is directly proportional to the moments of inertia of the wheels. From this it follows that an aluminum wheel consumes less power in attaining its angular velocity than a wheel of the same mass distribution but of greater weight.

The ratio of the energy of forward motion to the energy of forward motion plus rotation for a wheel is as 1 to 2. With a pneumatic-tired wheel having an outside diameter of 38 in., it is obvious how exceedingly important it is that rim weight should be kept as low as possible, particularly on low-gear cars and trucks where the rim inertia forces are a large proportion of the whole inertia of the vehicle. When a car is accelerating, say from rest to 20 m.p.h., the wheels have to be accelerated from rest to 40 m.p.h.

An interesting phase of the advantage incident to a reduction of the unsprung weight of a truck is illustrated by the following example. A certain 2-ton truck, highly efficient in its performance and of recognized merit, has a total unsprung weight of about 2600 lb. and a sprung weight of 4900 lb. We may assume that the engineer who designed this truck considered a ratio of unsprung to sprung weight of 1 to 1.8 as giving a proper distribution of weight and that this ratio was ample for the class of work expected of the truck. By substituting lighter wheels and introducing lighter metal in the axle parts, a saving of 658 lb. in the wheels and 160 lb. in the axles was obtained. This reduced the unsprung weight to 1800 lb. Using the ratio just mentioned, 1 to 1.8, we can maintain the same proportional weight distribution if we have a sprung weight of 3400 lb. The total weight of the truck with the heavy wheels and axle when loaded to

capacity was 11,500 lb., while a total loaded weight of the lighter truck would be 9200 lb. Computing the energy necessary to accelerate the truck from rest to 10 m.p.h., we find that the truck equipped with the lighter wheels and axles showed a saving of 20 per cent over the one with the heavier wheel installation.

A further slight conservation of energy may also be considered due to the reduced tire friction, and it is quite obvious that a considerable tire economy will be realized when the total truck weight is reduced. It should further be noted that a truck designed to the lighter figures mentioned, in which a reduction of 20 per cent in the power necessary for acceleration is attained, should show a materially better fuel economy.

AMERICAN ENGINEERING STANDARDS COMMITTEE

THE progress that is being made in the organization and proceedings of the American Engineering Standards Committee was outlined at the recent meeting of the National Bureau of Standards in Washington, by Prof. Comfort A. Adams who has been acting as the chairman of the committee and taken a deep interest in its establishment as a medium of supervisory cooperation in the formulation of national and international engineering standards.

Dr. P. G. Agnew, formerly of the Bureau of Standards, has become the secretary of the committee and commenced work in its office in the Engineering Societies Building at New York City. The chairman of the committee at this time is A. A. Stevenson.

It has been announced that the activities of the committee will be conducted according to a broad and generous policy. The committee expects to become the representative of this country in international standardization work. Five conferences in this connection are now in contemplation. A meeting on international pipe-thread standards was held at Paris, France, on Jan. 12, the American Society of Mechanical Engineers and the American Gas Association being the joint sponsors for this country under the methods of proceedure of the American Engineering Standards Committee.

The Society of Automotive Engineers and the American Society of Mechanical Engineers will form sectional committees on screw threads and ball bearings with international standardization in prospect. Additional subjects to be taken up by sectional committees are machine tool standards and limit gages. It is not unlikely that a sectional committee will be formed to consider the creation of industrial safety codes. One interesting point in this connection is the opinion which has been expressed that where labor organizations are interested in certain items of standardization bearing upon the safety of operators, they should be given committee representation.

A thought which has been expressed by Professor Adams is that the engineer stands between labor and capital, potentially at least, and that he should assume and maintain his status accordingly.

The general idea as to eligibility for membership on the American Engineering Standards Committee is that all organizations of national scope should be represented, including industrial organizations. A main purpose of the committee is to pass on the representativeness of standardization work conducted in the various engineering fields. The American Engineering Standards Committee will not endeavor to criticise the engineering standards established by competent bodies. It is expected that the Society of Automotive Engineers will take membership in the American Engineeriing Standards Committee.

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Tendencies in Engine Design

By L. H. Pomeroy[1] (*Member*)

Annual Meeting Paper *Illustrated with* Photograph and Drawings

At the moment, engine design is developing in the light of experience gained in the past five years of war, and according to the economic demands resulting from the world shortage of materials arising from the same cause. From the engineering point of view, the war turned half the world into a gigantic experimental shop in which the result was the only thing that counted, without regard to the cost of obtaining it. All services demanded that gasoline engines should be absolutely reliable in minor as well as major details of construction. Trucks and cars, for instance, had to be operated virtually independently of a service station, and under conditions where life might be endangered as much by a defective spark-plug as by a broken crankshaft. This imperative demand for absolute reliability showed very clearly the importance of detail design and manufacture. Cars and particularly trucks that were defective in some more or less unimportant detail for ordinary service were discarded during the war for these slight defects.

Second to reliability in importance came lightness of construction, either absolute or in relation to thermal efficiency, as in aeronautic engines, and of equal importance was thermal efficiency *per se*, with the capacity to use low-grade fuels. It is not suggested that all of these points were of equal importance on all kinds of gasoline engines, but some applied to all engines, and all applied to some.

The scope of the gasoline engine during the war was so wide, the necessity for its use so imperative, that engineers have been literally forced toward the solution of problems hitherto unrealized. Arising from these developments a vast amount of information is available, which he who runs may read. Airplane, touring car, truck, tractor and tank have been developed or stimulated by the test of war. The history of previous civilization, in which progress would have been delayed by thousands of years had a League of Nations been established 1000 years ago, repeats itself so that today we are in a position to take stock of the accumulation of war experience for the betterment of the gasoline engines of tomorrow. The engineer engaged upon a new design has a wealth of knowledge at his disposal, so that, although there have been but few variations from type due to the war, it is possible for him to design with a sureness of touch and to achieve results out of all proportion to the apparent difference between a new design and that of a few years ago.

The factors governing thermodynamic performance in respect to mean effective pressure, compression ratio and the effect of volumetric efficiency, have been recognized and known for many years, but the quantitative nature of these factors in gasoline engines running at high speeds has been only recently determined. Similarly, there have been many contributions to our knowledge of the mechanical performance of gasoline engines in respect to mechanical efficiency and internal friction, but the quantitative nature of the frictional effects arising from each organ is a product of recent research. Engine balancing has also been the subject of much attention, as engine speeds have been gradually raised to enable the high-speed characteristics of the gasoline engine to be fully exploited.

In general, the result of the past five years of intensive gasoline engine application has brought a host of problems under critical survey for the first time, and of no less importance is the re-examination of the position in respect to many designers' articles of faith that are crystallized in existing practice. It is, therefore, proposed to discuss the trend of design in the light of recent developments of the following:

(1) The effect toward obtaining absolute mechanical reliability
(2) The application of modern views of gasoline engine thermodynamics
(3) Improvements in mechanical efficiency
(4) Improvements in engine balancing
(5) Improvements in carburetion and the capacity to deal with low-grade fuels
(6) The principles of light-engine design
(7) Cleanness of design

Mechanical Reliability

The aim of the engineer during all time has been to overcome the weakness of the materials he has been forced to use and the defects arising from his own ignorance and carelessness, to say nothing of the ignorance and carelessness of others. The advent of the automobile, its immediate appeal to a pleasure-loving public and its enormous convenience and economic application, created a demand which it is not unfair to say killed or seriously wounded for some time the ideals of the engineer. The truth that the best is the cheapest, which has been demonstrated over and over in other engineering fields, was temporarily lost sight of and good engineering was subordinated to the whims of the purchase, production and sales departments. The result is well known. Many firms in both Europe and America have had to clean house and set out to incorporate good engineering, apparently at the expense of production, to find ultimately that accuracy and quality of machine workmanship were not only compatible with cheap production but reduced production cost, and that a few hours spent on the drawing-board were repaid infinitely by shop economies. The combination of good design, workmanship and material, reflects itself immediately in the satisfaction of the purchaser and the reduction of service charges and maintenance. It is safe to say that at no time in the history of the development of the automobile have the services and possibilities of the engineer been as widely realized by commercial administration as today. The underlying reason is, as usual, a matter of dollars.

The proportion of the catalog price chargeable to brains and labor is but a small one, at the outside not in excess of 20 per cent; the item chargeable to materials is at least twice this, and further, the difference between the cheapest and most expensive material is a still smaller

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percentage of the whole. The remaining 40 per cent or so of the total is sales cost and profit.

If there is one thing certain, it is that a bad car is not salable very long, even by the most skilled sales department; and, if we could imagine such a thing as a perfect car, the probable increased cost of labor, material and factory cost generally, would be amply saved in a reduction of sales charges. While the latter are something like twice the amount paid for the whole of the services used in producing the car, the argument is demonstrably fallacious that reduction of cost by reduction of quality is sound commercially. It is, therefore, reasonable and economical that, in the attainment of a high degree of mechanical reliability, the engineer should not be unduly hampered in respect to the cost of production and should be given free scope for living up to the tradition of being able, not only to make for $1 what any fool can make for $2, but to produce a result which allows any fool to sell for $2 that which previously only a spellbinder could sell for $1.

I have had the honor and pleasure of being conducted over many American factories during my stay in the United States and have seen with great pleasure the vindication of principles I have held for many years in relation to cost of production. By sheer manufacturing skill and enterprise, work of the first quality is produced, complex parts seem to have no terror for the production department and any rational increase in the complexity of design does not appear to be a consideration in arriving at a desired result. The engineer is far less limited here than in Europe in attaining his ends, and for this reason the goal of complete mechanical reliability is well within sight when an automobile will be as reliable as a watch.

What, then, are the lessons to be learned from the past few years in this respect? Possibly the first is that of the importance of material, and particularly the correct heat-treatment of steel. Aeronautic engine development has shown what can be done in these respects. The gasoline engine may be considered structurally as a combination of bearings and supports therefor. A large percentage of mechanical troubles are due to failure or wear of the bearings and distortion or fracture of their supports. The importance of choice of material in respect to bearing surfaces is almost as important as the lubrication system upon which they depend for their successful working, while rigidity of construction is a function of the supports and the correct geometrical disposition of the material thereof. As an example, piston-pin design can be taken. The older type of piston-pin for an engine of say 4-in. bore had a projected area of about 7/8 by 2 in. and gave trouble even then. The type of piston-pin in use on the Ricardo tank engines of 5⅝-in. bore had a projected area of 2 sq. in., an intensity of loading approximately two and one-half times that stated above and gave absolutely no trouble. This result was obtained primarily by a rigid support of the piston-pin and by first-class case-hardening.

Allied to this aspect of reliability has been the recognition of the great weakness imposed upon mechanical details through sudden variations of section and consequent mal-distribution of stress. Professor Coker, of London University, has evolved a means of visualizing by polarized light the stress distribution in transparent models of loaded members. Fig. 1 illustrates the stress distribution in a transparent model of a crankshaft, showing the high intensity of stress in the inside crankshaft fillets compared with the average stress and the imperative necessity for considering such local distribution of stress in parts of small dimensions as are commonly used in gasoline engines. The two curves shown are a measure of the stresses produced by a bending moment of one and two units respectively, the perpendicular distances from the contour of the crankshaft to the curves being a measure of the stress produced.

In 1914 I was largely responsible for the design of some very high-speed engines, running up to 4000 r.p.m., in which the crankshafts were to all appearance of ample stiffness and strength. Experience showed, however, that it was not until the crankshaft fillets were increased from 3/16 to ⅜-in. radius that reliability was attained. Fig. 2 illustrates the method in which the ⅜-in. radius fillet was obtained on crankshafts already machined with the orthodox 3/16-in. fillets. Probably 75 per cent of mechanical failures arise through a lack of appreciation of the stress distribution and the cultivation of a keen eye in this respect is of inestimable value; much more so than the capacity to use mathematical analysis, important as this is in the determination of general stresses.

Passing from structural reliability pure and simple, or the ability of the engine to withstand the forces set up within it, we come to the problems connected with temperature, fuel and lubrication. The last two are so interwoven that their relationship will be discussed in a later section. In respect to temperature problems it has been said that internal-combustion engine cylinder-design combines the problem of the hydraulic press with that of the boiler tube. It is necessary for a cylinder to withstand considerable pressure and also to be a first-class heat conductor. The one calls for thickness of metal, the other for thinness.

Fortunately, in gasoline engines of small size very little difficulty arises. The conductivity of the metal of which the cylinder is made is so overwhelmingly greater that the film resistance of gas and water on each side of it, that the temperature gradients in the metal are not sufficient to cause trouble. The heat resistance of cast iron ¼ in. thick is only about 1/100 of the resistance to the entrance of heat on the gas side and its emission on the water side. Hence, the disappointment arising in the use of aluminum for air-cooling cylinders, the advantage conferred by the high conductivity of aluminum being completely masked by the small proportion which the heat resistance of the cylinder wall itself bears to the whole. In the case of pistons and valves, however, the position is very different. In these the heat has to be conducted some considerable distance before it is dissipated, so that the conductivity of piston materials is very important. Principally for this reason the aluminum piston has found a ready demand. The advantages of aluminum pistons in relation to the reduction of reciprocating weight will be discussed later. Here it is sufficient to point out the great advantages of the aluminum piston from the point of view of the reduction of piston temperature. Some years ago Professor Hopkinson made experiments to determine the temperature of cast-iron pistons in a Siddely engine and found that this approximated 500 deg. cent. (932 deg. fahr.) at full load. Today the temperature of an aluminum piston of 7-in. bore at 1200 r.p.m. does not average over 200 deg. cent. (392 deg. fahr.). This achievement sets free possibilities for increasing the speed of large engines and of literally revolutionizing the design of the moderate-sized heavy internal-combustion engine on the one hand, and of allowing a still further increase in the speed of revolution of the small engine on the other.

Engine speed is no longer limited by considerations of piston temperature. For the same reason the designer can adopt large-bore cylinders with impunity if he so desires. As an example of progress in this direction to date may be cited a British six-cylinder engine developing 700 hp. at 1200 r.p.m. from cylinders of 7-in. bore and 11-in. stroke.

Although not related to temperature problems, the mention of aluminum calls for a remark upon the possibilities arising from the forged aluminum connecting-rod. Just as the aluminum piston permits an indefinite increase of piston size and revolution speed from the thermal point of view, so the aluminum connecting-rod removes another limitation of engine speed, that due to bearing pressure caused by the inertia effects of the big end of the connecting-rod and the inherent unreliability of a soft bearing metal under heavy loading. In my opinion the forged aluminum connecting-rod marks one of the greatest advances in the detail improvement of the gasoline engine and will have a very marked effect upon the trend of design, by no means limited to the connecting-rod itself.

Reverting to temperature problems, mention must be made of the advances in the materials from which valves are made. The use of aluminum for valves has not been developed commercially, although the aluminum valve is within sight. On the other hand, steels have been developed, notably nickel-chrome, cobalt and tungsten alloys, which successfuly resist the temperatures to which they are subjected. A further development has been that of

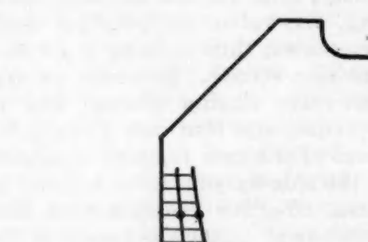

FIG. 1

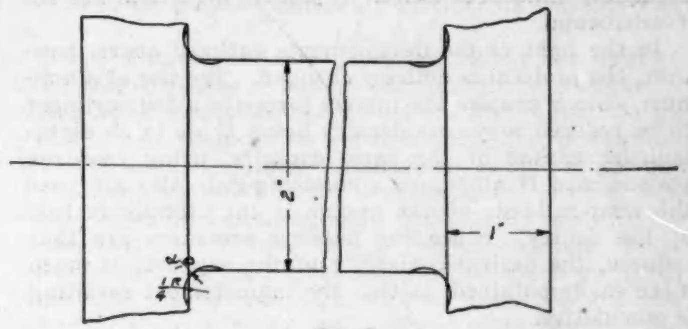

FIG. 2

the water-cooled exhaust valve, the cooling effect being obtained by drilling the valve, filling it with water and permanently plugging it. This practice is the invention of Sir Henry Fowler of the Royal Aircraft Factory in England. Although the evidence in support of the claims made in respect to a reduction of exhaust-valve temperature seems incontrovertible, the reason therefor is not easy to see. It seems inconceivable that the introduction of water, in itself an exceedingly bad conductor of heat, in place of steel can aid the conduction of heat from the head to the valve-stem. It also seems inconceivable that any convection effects which may be set up in the water can be as efficacious as direct metallic conduction, from the point of view of heat transference. It is obvious that the high specific and latent-heat properties of water can exercise their influence in heat absorption for only a short period. The authority behind the device, however, and the experimental evidence of its utility call for its description.

The various factors so far mentioned, namely, materials, stress distribution, rigidity of bearing supports, the use of aluminum pistons and connecting-rods and heat-resisting steels, are to be regarded as affecting reliability not only in the light of their application to existing designs, but also as to their influence upon the broad principles of gasoline engine design and construction. The ability of any engine to resist wear-and-tear is largely a function of the speed at which it is run. I showed many years ago that an engine would wear out far more quickly if driven by a belt at say 1500 r.p.m. than if it ran at this speed under load. Reliability apart from heat considerations is, therefore, largely a question of resistance to inertia effects, the working fluid pressures being of secondary importance. From this point of view there is considerable justification for the high-speed multi-cylinder engine of the eight and twelve-cylinder types, on account of the reduced inertia effects arising from subdivision of the working units.

The secondary unbalanced forces, for example, in an eight-cylinder engine are but 1.4 times those of a four-cylinder engine with the same size and similarly designed cylinders. For a given stroke the horsepower at any given speed is proportional to the square of the bore, while the mass of the piston is proportional to some power between the square and cube of the bore. It follows that the inertia forces in a four-cylinder engine are considerably higher per square inch of cylinder area than in an eight-cylinder engine of the same capacity running at the same speed. The same reasoning applies to the valves and tappets, where the inertia effects are of first importance. Other things being equal, it is fair to say that there is a greater potential reliability in an eight than in a four-cylinder engine of the same capacity. The actual practical differences arise principally from

secondary influences caused by human error and are not fundamental.

In the light of the developments outlined above, however, the problem is entirely changed. The use of aluminum pistons enables the inertia forces in a four-cylinder to be reduced very considerably below those in an eight-cylinder engine of the same capacity, using cast-iron pistons, and if aluminum connecting-rods also are used the wear-and-tear on the engine is cut literally in half at the source. Since the bearing pressures are thus reduced, the desirable rigidity of the supports is much more easily obtained, so that the improvement resulting is cumulative.

The providential coincidence that the lightness of aluminum is accompanied by high thermal conductivity is in its way as far-reaching and remarkable as the fact that two of the most important discoveries in relation to the development of the automobile, namely, the pneumatic tire and aluminum, arrived at the same time in history and were utterly unrelated to each other.

The designer of today has, therefore, an enormous unexploited field for the application of his experience and ingenuity. He is far removed from the limitations of only a few years ago, and the next decade should be productive of results which are startling both technically and economically.

Modern Views of Gasoline Engine Thermodynamics

Insofar as basic principles are concerned, the science of thermodynamics in its relation to internal-combustion engines generally and gasoline engines in particular is where it was. There have, however, been important contributions to our knowledge of the mechanism of combustion and the working properties of a mixture of gasoline and air. Chief among these is the recognition by automobile engineers of the phenomena associated with turbulence in their effect upon the gasoline engine, the investigation of the limit to which compression can be raised and the ancillary problem connected with so-called detonation or "pinking."

In 1912, Sir Dugald Clerk showed diagrams at the meeting of the British Association, illustrating the vital importance of turbulence in relation to the speed of ignition and, therefore, to engine speed. One of his diagrams is reproduced in Fig. 3. These diagrams were taken by an optical indicator from an engine in which a trip-valve gear was fitted so that the working charge could be retained within the cylinder and alternately compressed and expanded for a few revolutions before being ignited. The slow rate of ignition after two expansion and three compression strokes is clearly shown in the diagrams, proving how essential it is that the gas shall be in a state of violent motion at the instant of ignition. This state arises in the ordinary gasoline engines from the high velocity of the mixture passing through the inlet valve. The generally accepted opinion for many years has been that so long as the gas velocity is not less than 130 ft. per sec. sufficient turbulence is produced. I consider that the specification of valve area in terms of gas velocity is distinctly misleading. The effect of gas velocity upon turbulence must obviously be related to the revolution speed, for the reason that, although the turbulence produced by a gas velocity of 130 ft. per sec. is ample for rapid ignition at 1000 r.p.m., it might be very inadequate at 2500 r.p.m. It is necessary to specify valve area rather as a percentage of cylinder volume, however much this may offend the exponents of dimensional theory.

Recent experiments seem to show, however, that there is considerable ignition delay at very high engine speeds with such gas velocities, and that it is important to stimulate turbulence by all possible means. By so doing three ends may be attained.

(1) Reduction of the tendency to detonate, due to non-homogeneity of the working fluid; in other words, more thorough mixing of air and gasoline vapor inside the cylinder
(2) The ability to increase engine speed and still maintain high mean effective pressures
(3) Higher thermal efficiency due to the ability to raise compressions and subdue after-burning

The chief factors involved, from the designer's point of view, are the shape of the combustion chamber and the size of the valve required to allow for high volumetric efficiency. It is essential that the cylinder be fully charged at high speeds, and also that the inlet-gas velocity be high enough to produce sufficient turbulence for complete and rapid combustion, conditions which it will be readily seen are incompatible.

Some two years ago I conducted an exhaustive series of experiments upon two engines of approximately 180 cu. in. capacity, one with overhead valves, the other with side-by-side valves, to obtain data upon these points.[3] Taking compression pressure as a measure of volumetric efficiency, I found that with inlet-gas velocities of 300 ft. per sec., as good volumetric efficiencies were obtained as at any lower velocity. This was due to the effect of delayed inlet-valve closing, the valves in question closing at 45 deg. after bottom center, thus causing a virtual shortening of the compression stroke. However, as experiments with earlier inlet-valve closing showed but very little improvement in torque, and that only at very low speeds, the practical aspect of the case remains unaltered. I also found that, with the side-by-side valve engine, the results in respect to mean effective pressure and thermal efficiency were far and away below the practical ideal which should be anticipated from the compression ratio used and the high volumetric efficiency obtained. In the case of the overhead-valve engine, the mean effective pressure was some 16 per cent higher than that of the side-by-side valve engine, for the same compression ratio, although the thermal efficiency was not greatly improved. The combustion chamber of the overhead-valve engine, however, was far from ideal in shape, due to the use of one large inlet valve and two exhaust valves, so that, presumably, through contact with extended cold combustion-chamber walls, a large and approximately equal percentage of the fuel supplied to each engine was unburned during the explosion stroke in each case, the higher mean effective pressure in the overhead-valve engine being caused by greater turbulence due to the improved inlet-valve location.

In the past, combustion-chamber shape has been regarded almost entirely from the viewpoint of the heat loss to the cooling water during the early part of the explosion stroke, and the experimental and theoretical work done on this subject has gone far to mask the real importance of combustion-chamber design in gasoline engines. Sir Dugald Clerk has shown that in gas engines approximately half the cooling loss occurs in the first 30 per cent of the stroke and the remainder in the last 70 per cent. Gibson and Walker have shown also in experiments with slow-speed engines that the total jacket-heat loss is 18 per cent of the fuel supplied. An analysis by Lanchester of the Gibson and Walker results supports

[3] See *Proceedings* of Institution of Automobile Engineers, 1918-1919.

his reasoning that part of this heat loss is constant and depends upon the absolute cylinder dimensions, while the remainder is a constant percentage of the total fuel supplied and is due mainly to internal radiation, the heat loss to the cooling water being less at high speeds than at low speeds at full load. The effect of this reduction in loss at high speeds upon thermal efficiency was, however, in the Gibson and Walker experiments only about 1 per cent, although the jacket loss at 250 r.p.m. was 16 per cent, against 20 per cent at 100 r.p.m. The reason for this is that a large part of the jacket loss occurs in the exhaust stroke and therefore does not affect thermal efficiency.

We have, therefore, in an ordinary gas engine a loss of about 9 per cent of the heat supplied in the first 30 per cent of the stroke, or that part in which the combustion-chamber shape is most important. This is so little as to render considerations of the combustion-chamber shape of secondary importance in respect to the jacket loss and has led more than one designer astray in his search for high mean effective pressures and thermal efficiencies in gasoline engines. In my view, the attitude of indifference of many designers to thermodynamic problems is because the facts of the gasoline engine do not square with the theories of the gas engine, and these are largely reconciled if the considerations already mentioned in respect to combustion-chamber design are kept in mind. Briefly, the factors governing combustion-chamber design are first, compact geometrical shape; and, second, provision for adequate valve area.

The strongest argument in favor of the overhead-valve engine is its ability to burn the maximum proportion of the fuel supplied, owing to the compact combustion-chamber inherent in this valve system, the superior orifice coefficiency of the overhead valve being freely granted as against that of the ordinary pocket-located inlet valve. There is reason to believe, however, that a side-by-side valve engine can be designed in which the turbulence effect is even greater than in an overhead-valve engine, so that the extremes may meet. My experiments convince me that for engines of 40 to 60 cu. in. per cylinder a pocket-located valve will allow for a combination of this turbulence without any sacrifice of volumetric efficiency up to 3000 r.p.m., which is high enough for the present. It is significant that, after ten years or more of intensive high-speed engine design, there is no fixed opinion upon valve location, and that the manufacturers with the greatest experience with overhead valves do not use them in touring-car engines. In this connection it must be remembered that the question of side-by-side valve versus overhead valve is not necessarily determined on the grounds of efficiency and power alone. The purchasing public, who, after all, are the final arbiters in the matter, will give short shrift to thermodynamic perfection if this is accompanied by personal discomfort. As the ability of an engine to behave itself perfectly at low engine speeds and to possess great flexibility is not the least of the qualities demanded by the user, it must be pointed out that compact combustion chambers, such as arise naturally in the design of overhead-valve engines, are prejudicial to sweetness of running, particularly at low speeds and part throttle. Not the least virtue in respect to the orthodox side-by-side valve engine is the fact that the film of gas in contact with the comparatively cold walls of the combustion chamber burns during the expansion stroke with a quasi-Diesel effect, so that an indicator diagram taken under these conditions has a rounded top. This largely accounts for the popularity of the side-by-side-valve engine in spite of the deliberate sacrifice of economy and power.

There is room for much experiment along the lines of a compact combustion chamber, combined with a design in which the preservation of the cold gas film is realized, so that the best compromise between flexibility, power and efficiency can be obtained. All that can be said at present is that both side-by-side and overhead-valve engines represent opposite extremes, each introducing undesirable characteristics. In the light of recent research, I am inclined to think that it is going to be much easier to make the side-by-side valve engine more efficient and powerful than to make the overhead-valve engine sufficiently flexible. However, the ultimate type may quite

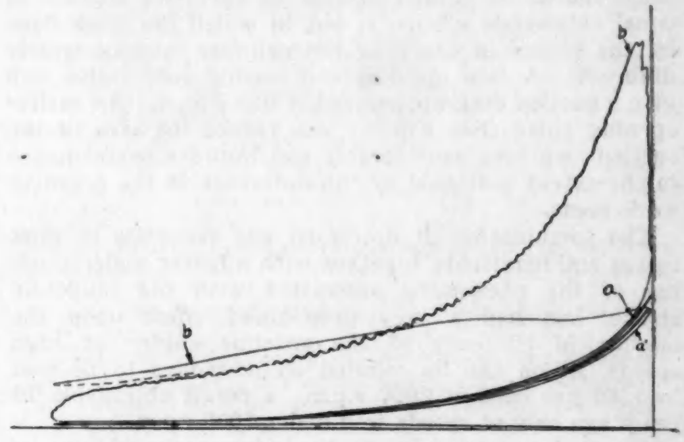

FIG. 3

conceivably be determined by secondary considerations, such as manufacturing convenience, silence and general appearance.

Improvements in Mechanical Efficiency

The mechanical efficiency of gasoline engines has been investigated by many workers and the records of engineering societies are full of data thereon. As far back as 1907, Professor Hopkinson at Cambridge University measured the mechanical efficiency of a four-cylinder Daimler engine by an optical indicator and one of his assistants, Mr. Morse, evolved the method that is so largely used today of measuring mechanical efficiency by cutting out one cylinder of a multi-cylinder engine and finding the torque exerted by the remaining three. The difference at the same speed between the torque of three cylinders and that of four in a four-cylinder engine is the indicated torque of the non-firing cylinder, thus allowing the relation between brake-horsepower and indicated-horsepower to be measured with four readings. The analysis of frictional losses of gas engines was also investigated by Hopkinson, and those of high-speed gasoline engines by Riedler in Germany, to mention only two of many investigators.

The overall mechanical efficiency of both gas and gasoline engines was found to be remarkably independent of their relative sizes, being about 85 per cent in large gas as well as in small multi-cylinder gasoline engines. Great falling off in the mechanical efficiency of the gasoline engine with increase in speed was, however, apparent.

In my opinion, it is most unfortunate that mechanical losses have to be subdivided into those arising from gas friction and bearing friction, but the definition is too well established to be worth contesting. The gas friction

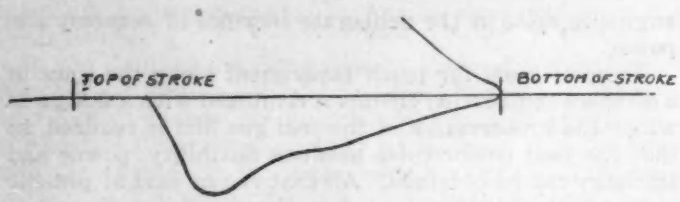

FIG. 4

or pumping losses may therefore be discussed first. These, it is well known, arise from the friction due to ports and valves, both intake and exhaust. In the normal gasoline engine the gas flow into the cylinder is due to the difference between the pressure in the cylinder and that of the atmosphere. It is also possible to have two engines of equal volumetric efficiency, but in which the work done by the piston in charging the cylinder may be widely different. A late opening and closing inlet valve will give a suction diagram somewhat like Fig. 4. An earlier opening valve (See Fig. 5) will reduce the area of the suction loop very considerably and improve performance to the extent indicated by the difference in the negative work areas.

The combination of decreased gas velocities in inlet valves and manifolds, together with a better understanding of the phenomena associated with the induction stroke, has had a very pronounced effect upon the mechanical efficiency of the gasoline engine at high speeds, which can be counted on nowadays to be well over 80 per cent at 2000 r.p.m., a result obtainable 10 years ago only at speeds well under 1000 r.p.m.

The back-pressure due to the exhaust should also be mentioned if only to point out its relative unimportance owing to the rapid escape of the exhaust gas when the exhaust valve is opened. Further, it is easy to arrange exhaust pipes so that there may actually be a negative pressure in the cylinder toward the end of the exhaust stroke. The all-important point is that exhaust gas under pressure is not trapped in the combustion chamber when the valve closes. With valve areas such that gas velocities do not exceed 300 ft. per sec., the influence of gas friction upon mechanical efficiency is not of much account, nor is it susceptible of much reduction. The frictional resistance of the carbureter itself is usually far in excess of the inlet port and valve friction in the practical case and has to be tolerated to get efficient carburetion.

Leaving the question of fluid friction losses and passing to those more purely mechanical, these may be divided as follows:

(1) Losses due to driving auxiliaries, such as the fan, the water-pump, the oil-pump and the electrical equipment
(2) Losses due to bearing friction of the crankshaft, the camshaft, the valves, the gears, etc.
(3) Piston friction

The losses due to driving the water and oil pumps and electrical equipment are neglible in most cases. The loss

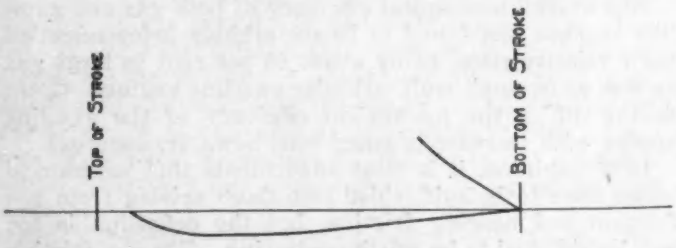

FIG. 5

due to driving the fan, particularly at high speeds, is however considerable, easily amounting to from 2 to 3 hp. at 2500 r.p.m. It is noteworthy that the vast majority of designers employ gears to drive the water-pumps which are often larger than those in the transmission, but are quite content to drive the fan by a belt, in spite of the fact that the fan absorbs about forty times the power required to drive the pump.

It has been known for many years that the friction of a well-designed journal bearing is exceedingly small, but it seems to have taken a long time to make the logical deduction that if the journal friction in a gasoline engine is small and the total friction absorbs some 10 per cent of the indicated-horsepower, a large proportion of this must be charged to friction between the piston and the

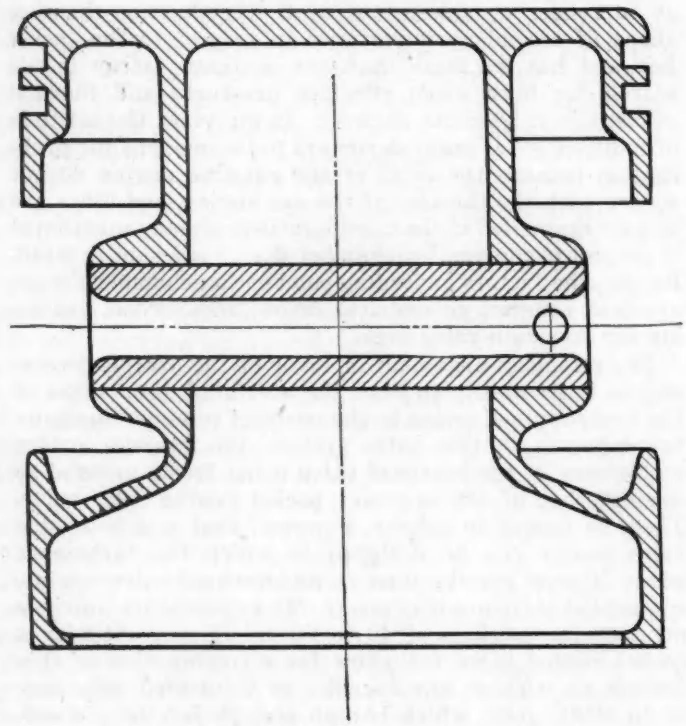

FIG. 6

cylinder walls. It has taken still longer to seriously attack this problem of reducing piston friction.

In gasoline engines with cast-iron trunk pistons, the piston friction is about 8 lb. per sq. in. of piston area at a piston speed of 2000 ft. per min. The magnitude of this quantity depends upon the piston thrust and rubbing velocity, the former item being very largely a function of the inertia pressure. Reduction of piston mass is obviously the first step in reducing piston friction. The advantages of the aluminum piston in this respect are apparent. It is not generally appreciated that the prime virtue of light reciprocating parts is the improvement of mechanical efficiency caused thereby, and that, apart from this, the net energy absorbed in reciprocating a piston is the same for heavy pistons as light, being nil.

The next point of importance is the consideration of the action between the rubbing surfaces of the piston and the cylinder wall. In the case of a trunk piston of normal design it is easily seen that the area of the oil film between these surfaces is equal to the wall area of the piston, and that as only one side of the piston is functioning

at a time there is roughly four times more oil film being acted upon than is required for taking the piston thrust, which probably does not extend to more than one-quarter of the piston circumference. This unnecessary surface in contact is naturally prejudicial to mechanical efficiency. For this reason many pistons are relieved for part of their length by reducing the diameter about the piston-pin and are also drilled to reduce the surface in contact. The advantages obtained are real and distinctly shown on the test bench. Unfortunately, it has been found dif-

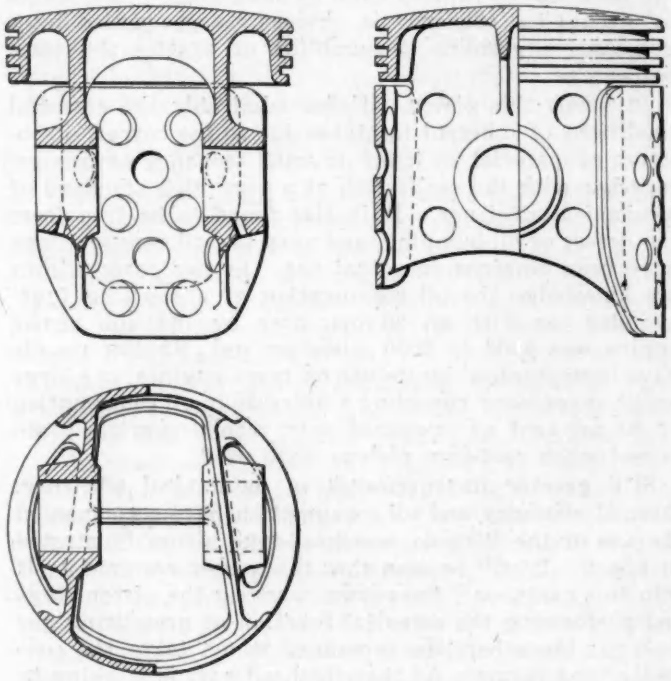

FIG. 7

ficult to combine mechanical strength with lightness in these pistons and many failures have resulted from such attempts to side-track the defects of the piston of standard design. Further, the disposition of the bearing surface in pistons so relieved is not correct. There is still a 50 per cent excess of unnecessary surface, allowing for the fact that bearing surfaces have to be provided on each side of the piston, although only one side functions at a time. The rigidity of the piston-pin bosses is also seriously reduced unless the piston walls are unduly heavy. The orthodox trunk piston has been regarded by designers in much the same way as the flatness of the earth was regarded by the world in general before the voyage of the first American immigrant. It has simply been accepted without question. The success attending the various departures from orthodox construction is a very useful object lesson that no established construction is beyond improvement.

One of the earliest of these departures is the Zephyr piston shown in Fig. 6. This type of piston has been very successful in aeronautic and racing engines, some of the first specimens being used by me on racing cars in 1912. In addition to the reduced bearing surface and adequate piston-pin boss support, it will be noticed that the design of the piston crown is well adapted to dissipating heat. A still further development toward a rational solution of the piston problem is that of the Ricardo slipper piston. This is shown in Figs. 7 and 8. It will be seen that there are a considerable number of

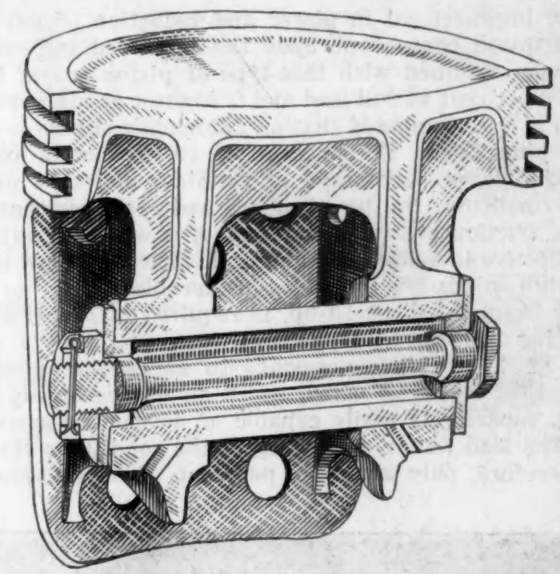

FIG. 8

interesting points in this design, which can be tabulated as follows:

(1) The direct transmission of piston thrust to the slippers
(2) The proportioning of the slippers to the loads they carry, the compression slipper being reduced in area compared with that receiving the explosion thrust
(3) The slippers extend the whole working length of the piston and only laterally to the degree required
(4) Rigid support of the piston-pin bosses
(5) The ability to use a floating piston-pin
(6) Inherent lightness

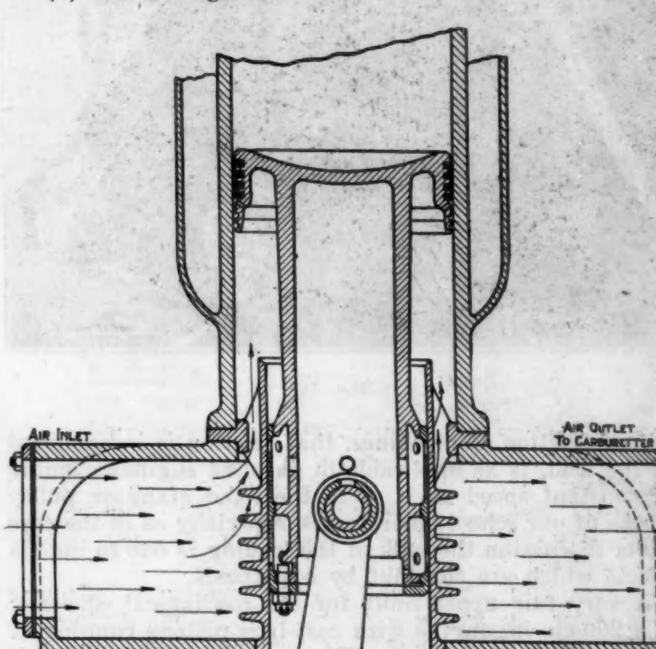

FIG. 9

The improvement in power and reduction of gasoline consumption consequent upon the improved mechanical efficiency obtained with this type of piston varies from 5 to 10 per cent at full load and is highest at high speeds. When it is remembered that an automobile engine is only developing about 25 per cent of its maximum power during a large percentage of running, and that under these conditions the inertia force and speed elements of piston friction are fully manifested, it will be seen that the effective increase in mechanical efficiency at low loads and high speeds may be very considerable.

For example, about 10 hp. is required to propel a car weighing 3000 lb. at a speed of 30 m.p.h., with a gear ratio of 5.01; this corresponds to an engine speed of about 1550 r.p.m. at which speed an engine of say 200 cu. in. capacity is easily capable of developing some 36 hp. The load factor, according to the conditions stated, is, therefore, only about 28 per cent. The well-known

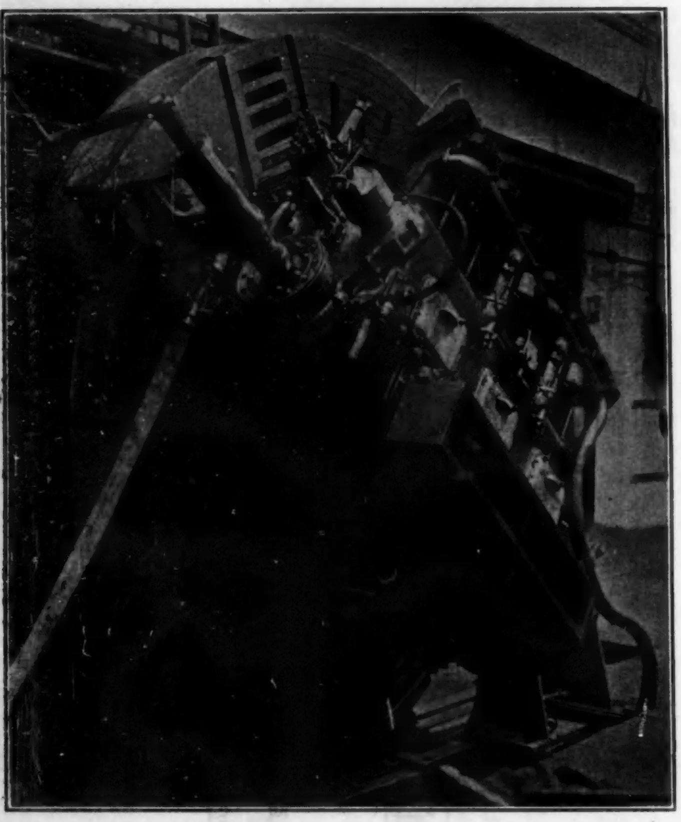

FIG. 10

law of friction of machines, that friction is independent of the load, is as applicable to gasoline engines running at constant speed as it is to the hand crane or pulley blocks of our school laboratories, especially as in the case under discussion the bulk of the loading is due to inertia effects which are constant by hypothesis.

A very fair upper limit for the mechanical efficiency of a 200-cu. in. engine with cast-iron pistons running at 1500 r.p.m. is 87 per cent. The corresponding figure with slipper-type pistons may easily be 92 per cent. The friction-horsepower for cast-iron pistons is, therefore, $\frac{36}{87} - 36 = 5.2$ hp., and for the slipper pistons $\frac{36}{92} - 36 = 3.2$ hp.

As this friction loss is virtually independent of the load, it will be the same when the engine is delivering the 10 hp. at 1500 r.p.m. as when delivering the maximum horsepower of 36 at this speed. The mechanical efficiency with cast-iron pistons, the engine delivering 10 hp., will, therefore, be 66 per cent, while with the slipper piston the corresponding mechanical efficiency will be 76 per cent, an improvement of over 15 per cent at 28 per cent of full load, arising from an assumed improvement of 5 per cent at full load. In actual practice, the improvement at full load is usually far more than 5 per cent, as few engines with cast-iron pistons have a mechanical efficiency of 87 per cent, thus enhancing the corresponding item under ordinary running conditions. The increase in mechanical efficiency is obviously accompanied by a decrease in gasoline consumption of exactly the same percentage.

In brief, this piston satisfies admirably the essential conditions of inherent lightness due to the correct disposition of material, in itself no small economic advantage, together with the realization of a very high standard of mechanical efficiency. It is also found to be free from the defect of oil-pumping, and very low oil consumptions have been obtained in actual use. In two cases within my knowledge the oil consumption of a sporting four-cylinder car with an 80-mm. bore by 180-mm. stroke engine was 4000 to 5000 miles per gal. Similar results have been obtained by its use on truck engines, one large omnibus company recording a decrease in oil consumption of 80 per cent as compared with results previously obtained when cast-iron pistons were used.

Still greater improvements in mechanical efficiency, thermal efficiency and oil consumption have accompanied the use of the Ricardo crosshead-type piston illustrated in Fig. 9. It will be seen that this piston resolves itself into two parts, one, the crown carrying the piston-rings and performing the essential function of preventing gas leakage; the other, the crosshead which takes the connecting-rod thrust. As the crosshead part is working in a guide that is kept cool by the flow past it of the air on its way to the carbureter, and is also ideally lubricated, the working clearance can be reduced to very fine limits so that the piston is absolutely silent. It further insulates the crankcase from the heat radiated from the piston, thus keeping the lubricating oil cool. The supply of oil to the cylinder walls can be precisely regulated and is exceedingly small in quantity, as the top of the piston does not perform any thrust-resisting functions. Further, the crankcase can be filled to the level of the crankshaft, without increasing the amount passing to the cylinder walls. The net result is a very low oil consumption of 0.01 pint per b.hp.-hr. in a 200-hp. engine at full load. There is also an almost complete absence of carbonization. The official report of a British Government department on an engine with this type of piston stated that at the end of a 400-hr. run the carbon could be rubbed off the piston by the fingers.

It is also a most interesting and important characteristic of engines with this type of piston that while at full load the temperature of the air in the region of the crosshead guide is about 70 deg. fahr. above atmospheric temperature, falling to that point before entering the cylinders, at light load the respective temperatures are about 90 and 30 deg. fahr. above atmospheric temperature, thus giving the ideal conditions for prevention of gasoline condensation at low loads and speeds, and when idling. The use of heated manifolds, hot spots, etc., is thus rendered unnecessary.

Fig. 10 shows a tank engine fitted with these pistons. In this position the engine was required to run light

for 30 min. and then to open up immediately without smoking.

In the foregoing remarks on mechanical efficiency adequate lubrication has been assumed. The relative merits of the various systems in use do not seem to have changed much and it is impossible to say that any particular one is greatly superior to the others. The system that is adopted seems to be largely a matter of taste, which indicates the verification by experience of the doctrine that a correctly-designed bearing will lubricate itself if oil is led to it.

Under no circumstances is it likely, in gasoline engines, that a bearing can be oil-borne by the pressure in the oiling system. On the other hand, the advantage of forced lubrication in cooling heavily-loaded bearings by sheer volume of oil pumped is not to be lightly set aside.

Improvements in Engine Balancing

This aspect of engine design would more properly be termed improvement in means for reducing engine vibration, since vibration is by no means eliminated when complete balance is obtained, a fact well known to most designers of six-cylinder and even twelve-cylinder engines.

In respect to engine balance *per se*, the position to date is that engines with two, four, six, eight or twelve cylinders, are or can be completely balanced in respect to primary and secondary unbalanced forces and couples. The six and twelve-cylinder engines are of course inherently balanced; the four and eight-cylinder need the application of a device for neutralizing the secondary unbalanced forces. Fig. 11 shows the Lanchester anti-vibrator, and Fig. 12 the Ricardo secondary balancing device, each if which is effective. The principle of the Lanchester device is that of two reverse-rotating bob-weights that apply equal and opposite forces to the secondary inertia forces set up by pistons at the end of each stroke, the bob-weights rotating at twice the engine speed. As the energy content of the bob-weight system is constant at a constant speed, the only force required to drive the device is that arising from the friction of the bob-weight spindles, so that the driving mechanism is merely a motion transmitter. When the engine is accelerated or decelerated, tooth pressures of material magnitude arise, but can be dealt with easily. The peripheral speeds of the gears is 90 ft. per sec. at an engine speed of 3000 r.p.m., approximately the same as with many turbine reduction-gears in daily use, where, in addition, the tooth pressures are exceedingly high.

The Ricardo device is based upon the principle of introducing reciprocating masses driven by linkages, producing the same angular effects in respect to inertia with these masses as the connecting-rod crank system produces with the pistons. It is exceedingly ingenious. The conditions under which the pin-joints in the linkage systems work are no worse than those of the piston-pin, and although the device appears a little complicated, it is simple in detail and works well. It has not, however, the advantage of ready application to existing designs possessed by the Lanchester anti-vibrator. I have corresponded with many people on the subject of secondary balancing and have found a general vagueness of thought thereon, many confusing secondary balancing with crankshaft torsional damping and still more stating that they had attained the same ends by crankshaft counterbalance-weights. Whatever effect counterbalancing may have in reducing bearing loads, it certainly does not, *per se*, affect the balancing problem one way or the other. The only virtue of crankshaft counterbalancing, the addition of

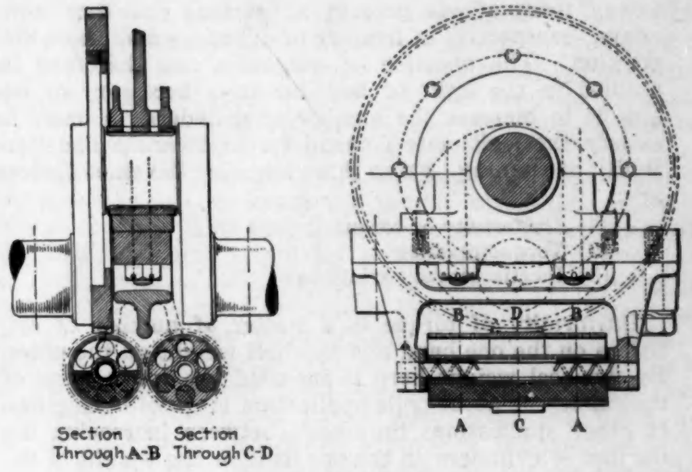

Fig. 11

balance weights to the crank webs so that each individual crank and attached rotating masses may be balanced, is that wear on the middle main-bearing of the crankcase can thus be practically eliminated, and the crankcase itself reduced in weight very considerably.

The whipping tendency or "skipping-rope action" of an unbalanced crankshaft, particularly in a six-cylinder engine, sets up very heavy bearing loads and crankcase stresses, so that counterbalancing may be very desirable in engines with center crankshaft bearings. The elimination of the center bearing, another accepted fetish, dodges the bearing loading problem very well, and will doubtless be more widely adopted than at present. Further, the use of counterbalance-weights should be accompanied by an increase in crankshaft diameter, owing to the torsional mass effects of the balance-weights which may easily exemplify the adage that the cure is often worse than the disease.

Apart from the question of structural rigidity, recipro-

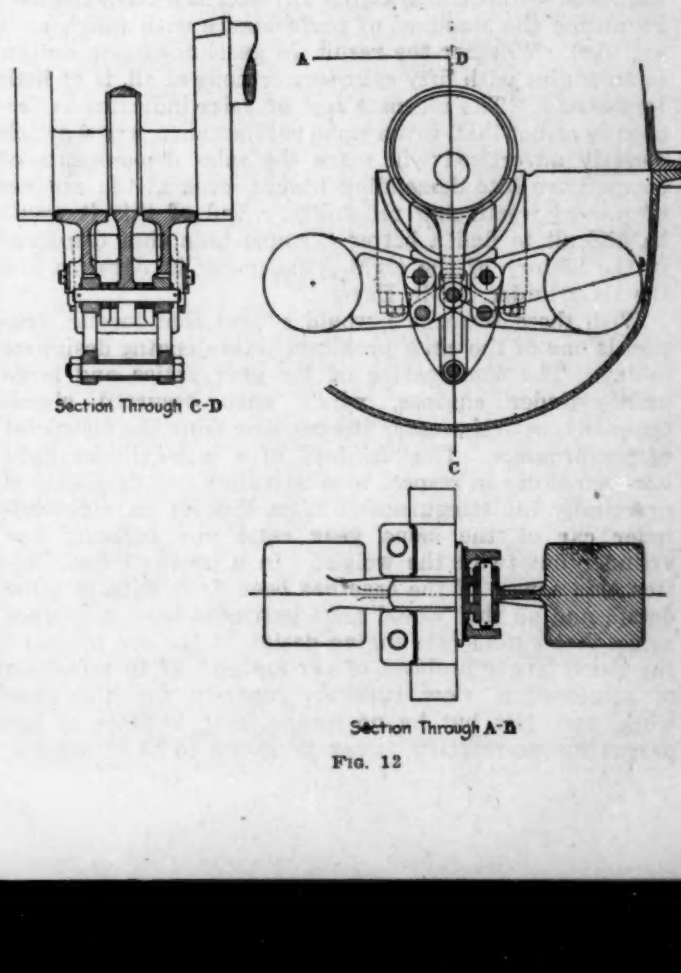

Fig. 12

cating mass effects present no serious problems nowadays, irrespective of number of cylinders and crank disposition. The question of vibration can therefore be studied in the light of how far it is necessary or desirable to increase the number of cylinders, in order to satisfy the legitimate demand for smoothness and flexibility of running. From this viewpoint the chief factors are:

(1) Uniformity of torque
(2) Torque reaction
(3) Inertia torque variations

Uniformity of torque is a matter of number of cylinders on the one hand and flywheel weight on the other. By flywheel weight here is included the whole mass of the car in the automobile application of gasoline engines. In other applications the choice between increasing the number of cylinders on the one hand or the weight of the flywheel on the other would be a matter for purely individual taste, if the problem had no other aspect. In either instance, sufficient torque uniformity can be obtained to satisfy the most exigent case.

The question of torque reaction is not disposed of so easily. It is in respect to the reactive effects upon the crankcase supports, caused by the fluid pressure in the cylinders, that 75 per cent of the argument lies in the choice of the desirable number of cylinders, particularly for automobile use. In a discussion of this kind the point that the purchaser wants a six, an eight or perchance a twelve-cylinder car and therefore must be so supplied is quite irrelevant except insofar as it calls for an analysis of the situation in general. Whatever the reason for the public taste with regard to multi-cylinder engines, whether it be actuated by advertising or genuine need, the outstanding fact is that the multi-cylinder engine has a large following, and therefore gives the purchaser something which appeals to his sense of comfort and fitness.

It is the business of engineers to translate the often inarticulate but nevertheless real desires of the user into mechanical terms, and to consider how design can be improved without demanding any sacrifice from the user regarding the standard of performance with which he is satisfied. Whether the result, in gasoline-engine design, is an engine with fifty cylinders or none at all, is of little importance. The ultimate test of sales indicates as frequently as not that, given equal performance, a new article cleverly advertised will scare the sales departments of competitors into demanding novelty even at the expense of proved worth and reliability. And of this it would be difficult to find a better example than that displayed in the history of the eight-cylinder car in America and the six-cylinder car in Europe.

With these remarks I would submit that torque reaction is one of the chief problems before engine designers to-day. The combination of low-gear ratios and large multi-cylinder engines, which have occurred simultaneously, entirely masks the problem from the viewpoint of performance. The "feeling" of a four-cylinder light low-geared car in respect to acceleration and flexibility is practically indistinguishable from that of an eight-cylinder car of the same gear ratio and capacity per cylinder but twice the weight. In a previous contribution this aspect of the case has been dealt with in some detail, and all that would seem pertinent here is to urge again that automobile engine design is but one phase of the much larger problem of car design. If in reference to acceleration, slow running, capacity for high-gear work, and, last but by no means least, absence of apparent torque reaction, it can be shown to be reasonable that the attainment of an accepted standard is primarily a function of car weight, the number of cylinders which will fill the conditions reduces itself to a matter of simple arithmetic.

It is quite conceivable that the four and the eight-cylinder engines may win out, each in their respective spheres. They are natural relations and between them they can certainly fill any existing demand. Unfortunately the eight-cylinder engine as now designed is very wide and precludes in some cases an artistic hood and radiator design. But this is by no means inherent in the type, as a later section of the paper will indicate. Briefly, the case for appearance in relation to the eight-cylinder engine rests upon the short-stroke engine with short connecting-rods and secondary balancing, and an "eight" designed from this viewpoint should present many attractive features.

In the list of the factors contributing to vibration disturbance, mention was made of the torque variations due to piston inertia. This subject is but little appreciated and a few words may be of interest. This torque variation arises from the fact that the pistons are brought to rest twice per revolution, thus causing alternating tension and compression forces in the connecting-rod. The horizontal component of these forces evidently tends to rotate the crankcase about the crankshaft axis and causes a variable torque reaction upon the crankcase supports. Under certain conditions these reactions are neutralized by the fluid pressure in the cylinder. A little thought will show that if at the latter part of the compression stroke the connecting-rod is in tension due to piston inertia, this tension can be neutralized, or in fact altered to compression, by the fluid pressure upon the piston due to compression. Under these circumstances the torque reaction due to fluid and inertia pressure will be zero. There is clearly a field of engine speed and compression pressure in which the compression will largely neutralize the torque variation due to inertia.

As the compression pressure may be considered constant with an increase of speed and the inertia effects vary as the square of the speed, it is obvious that the inertia effects will start by being less than the compression effects and gradually eclipse and finally exceed these. This inertia torque variation is often unpleasantly manifested when the throttle is suddenly closed at high speeds, and the cushioning effect of the fluid pressure removed. Many engines then behave as if a bag of nails had been introduced into the crankcase, and at least one patent has been filed relating to carbureters in which the supply of gasoline but not air is cut off when the throttle is suddenly closed. The fundamental remedy is to neutralize the inertia effects at the source.

Fuel and Carburetion

The problems arising from the deterioration in the quality of gasoline and the economic necessity of using low-grade fuels in gasoline engines, have been the subject of very extended study in America and been discussed far more authoritatively than is possible here. Fuel questions are indigenous to America, and find their natural solution among American engineers. There are, however, one or two points relating to the trend of engine design to which reference may be made.

The chief troubles arising from the use of present-day gasoline, the word low-grade being now superfluous, are:

(1) Detonation or pinking
(2) Prevention of gasoline deposition in the crankcase

Detonation may be considered as an instantaneous increase in pressure as distinguished from an explosion in which the time factor is marked and measurable. In a mixture of gasoline and air, it apparently depends upon the pressure and chemical constitution of the gasoline at the instant of ignition rather than upon the temperature. Fuels containing a proportion of the aromatic series or of benzol, are far less susceptible to detonation than those not so constituted. Detonation is also less likely to arise if the mixture is in a state of violent turbulence so that ignition can spread rapidly through the charge. If this is not the case, local ignition apparently sets up a compression wave that spreads more rapidly than the ignition so that when the ignition takes place in parts of the charge remote from the firing point, the charge is very highly compressed and therefore detonates.

The use of benzol in even small proportions of 10 to 20 per cent will allow much higher compressions to be used than are possible without such addition. The admixture of cooled exhaust-gas has the same effect and has been used for many years on producer-gas engines where very high compressions are common. The application of this idea to gasoline engines generally is now being worked out and will doubtless become common. The difficulties are purely of detail application and in the automobile engine largely arise through the necessity for part-throttle working. I have applied cooled exhaust-gas to a badly carbonized engine that simply would not run at full throttle with the result that full throttle could be used, the torque obtained being within 1 per cent of that given by the same engine when clean.

Many interesting data on this subject of detonation have been contributed by Mr. Kettering. His researches in this connection are eminently worthy of detail study. The curves he has obtained show clearly that detonation occurs well down on the firing stroke, which supports the view that it occurs through delayed local ignition of a highly compressed part of the charge.

The subject of gasoline deposition in the crankcase is one of more practical importance than a problem involving physical speculation. It has been known for many years that the lubricating oil in the crankcase gradually absorbed gasoline, and retained such fractions as were not volatilized at the temperature of the lubricating oil during use. With prewar gasoline, the maximum absorption was about 4 per cent. Nowadays, when fully one-third of standard gasoline does not vaporize under a temperature of 140 deg. cent. (284 deg. fahr.), deposition is much more serious. It can be dealt with once and for all by the use of the crosshead piston described above. On the other hand, this cannot be applied without radical alteration in design. The use of high-temperature induction pipes is therefore indicated to vaporize the gasoline-air mixture as much as possible before it enters the cylinders. The standard method of exhaust jacketing the induction manifold seems defective in that the heat supply is least at part throttle, when it is most required, and greatest at full throttle, when it is least required. The correct way seems to be to reverse this state of affairs. A device which would satisfactorily and automatically pass the greater part of the exhaust-gases through the manifold jacket at low throttle and only a small portion at full throttle appears to be desirable.

Such a device is illustrated in Fig. 13. It depends for its action upon the vacuum in the inlet manifold, which is more or less in accordance with the necessity for external heat supply. The spring-loaded piston is in communication on one side with the induction pipe and at the other forms a valve controlling the amount of exhaust-gas which can pass into the inlet-manifold jacket. At a small throttle opening, or a high vacuum in the induction pipe, the maximum amount of exhaust-gas is passed into the manifold jacket while at full throttle, or a low vacuum, the supply of exhaust-gas is cut off, thus avoiding unnecessary preheating of the mixture.

It seems probable that the maximum deposition of gasoline in the crankcase occurs when the engine is started. At this time the cylinder walls are covered with cold oil which is quickly scraped off by the rings and not replaced with further lubricant until the engine becomes warm. Further, the gasoline which is used during the first few minutes after starting is that which is left in the float-chamber after the previous run, from which the lighter fractions have already in all probability disappeared. The same condition holds for vacuum tanks, where the surface of the gasoline is exposed to the atmosphere. It was the custom among the drivers in the Mechanical Transport Department in France to drain the carbureter float-chamber before starting up and, as engine starters were not in common use, the practice may be accepted as one that was really

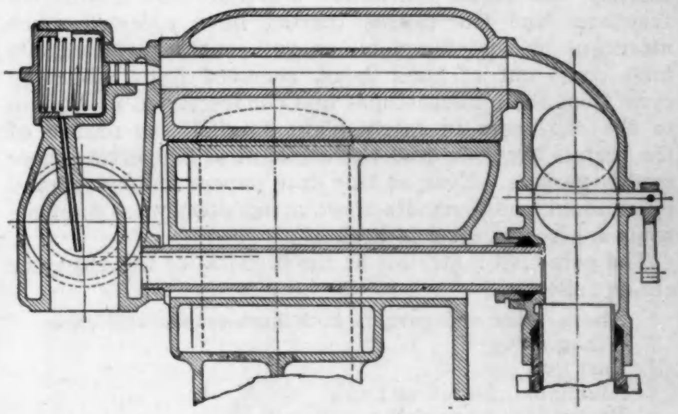

Fig. 13

necessary. Since the preheating of induction-pipe manifolds before starting is very advantageous, the draining of the float-chamber, if arranged for, as it can be, so that no troublesome operations are involved, will minimize the serious effects of gasoline deposition.

In respect to carburetion generally, the difficulty is primarily in securing proper distribution. It has been realized that it is much easier to produce a spray of gasoline and air in requisite proportions at all loads and speeds, than it is to insure that this spray is distributed equally in correct quantity and quality to the cylinders. There does not seem to have been much radical improvement in induction-manifold design, although there is less variation in this respect in engines than existed a few years ago. The principal detail improvement is possibly the reducing of the size of the manifold so that a high gas velocity is attained through it. There is a real need for an inlet manifold the area of which will adjust itself to the requirements of the moment, being large at full load and small at light load. A flexible lining of the manifold, with a space between it and the manifold open to the atmosphere, would seem to possess the property of collapsing at light loads when the vacuum in the manifold is high and of expanding at full load when the vacuum is low, thus producing an area proportional to the load. On the whole the situation is probably accurately summed up in the words of Mr. Howard Marmon to the author, "No deposition, no distribution problem."

Fig. 14 shows the fractional distillation curves obtained by G. H. Baillie, published in 1908 in a paper read before the Royal Automobile Club. These are of interest as recording the properties of gasoline of a decade ago, and indicate strikingly how the carburetion problem has changed in respect to the fuel itself.

The "war spirit" used in England during the last three years, when fractionally distilled, only evaporated some 70 per cent of its volume under 140 deg. cent. (284 deg. fahr.), while the 0.76 shell in Baillie's experiments was completely evaporated at this temperature.

Materials

It would be neglecting the subject of the trend of engine design not to deal somewhat fully with materials, and the principles underlying the choice of material for various purposes. The literature and practice of the past five years indicate that the physical properties of materials are considered far more intelligently than ever before and that design is based upon some knowledge of the real and intrinsic merits of the materials in use. The term "a stiff steel" is seldom heard now that it is more generally appreciated that all steels possess approximately the same stiffness. Further, the nature of fractures and the causes thereof have received much attention by metallurgists, or rather the metallurgists have come out of their lairs, removed their all-seeing eyes from their microscopes and condescended to explain to the engineers in fairly plain English the nature of the metals they use and the mechanical properties these metals possess. Even at this date papers are being read by eminent metallurgists speculating upon what happens when a piece of steel is broken.

The principal materials at the disposal of the engineer are as follows:

- Steels, alloy and carbon, both high-tensile and case-hardening
- Cast iron
- Aluminum, forged and cast
- Bronze, phosphor and manganese alloys
- Bearing metals containing lead, tin, copper and antimony

The choice of material for any specific part calls for considerable judgment to obtain the best results. It is perfectly easy to specify a high-tensile alloy steel for a crankshaft on the grounds that this is accepted high-class practice, although for automobile engine purposes, even of the highest class, this may be sheer waste of money. The average stress in a crankshaft of the solid type is very low, and does not justify high-tensile alloy steel except from one point of view. This is the local increase of stress in small fillets explained previously. As the intensity of stress in a fillet varies inversely as a high power of the radius, it would seem far more rational to eliminate these high local stresses by increasing the radius of the fillets, than to use a considerable weight of high-tensile material all through with attendant expense and difficulty of working. In all cases, subject to the above reservation, the dimensions of a crank are those required for stiffness rather than strength, and are largely in excess of stress requirements. The actual deflection under a given load of two crankshafts of identical design is strictly in proportion to the relative moduli of elasticity of the materials of which they are composed. These do not vary more than 2 or 3 per cent in any of the steels in common use, the higher values for stiffness being frequently obtained with the lower-tensile steels. For the above reasons there is little advantage in the use of expensive alloy steels for cranks unless their properties are essential to overcome bad design or are made necessary by the need for a reduction in section without sacrificing stiffness, as in bored-out aeronautic engine crankshafts. In fact, the use of carbon steel properly heat-treated is very much overlooked in engine construction generally. The advantage of alloy steel has largely disappeared with increasing knowledge of heat-treatment and stress distribution.

In respect to physical tests, the old-fashioned tensile test probably gives the best indication of the relative qualities of steels in general, in spite of the fashion to regard shock and fatigue tests as decisive. It is worthy of remark that the high-tensile steels invariably give much lower impact-test figures than low-carbon low-tensile steels. The real point is that impact and fatigue tests only convey information as to the capacity of the material to resist fracture, when stressed outside its elastic range. If such stress condition occur, fracture is bound to occur sooner or later, and if such is the case, the precise amount of energy absorbed during fracture does not possess much interest. It seems more important to have accurate information upon the true limit of proportionality of stress and strain, and in general much more information of the behavior of steel inside the so-called elastic range.

My practice in specifying and using steel has been to insist upon having the analysis of the steel within reasonable limits and to be sure, from micrographic examination, that the steel possesses the desired structure for the particular purpose intended. With these two points assured, physical properties will usually take care of themselves. An illustrative example of the futility of trying to combine high-tensile and high-elongation properties arose in England during the war in connection with aeronautic engine crankshafts, many perfectly sound cranks being rejected which would have been passed in the light of analysis and micrographic examination.

Of cast iron, little need be said except that machining and wearing properties are usually inconsistent with each other. It seems a pity to use soft cast iron which wears rapidly at the high piston-speeds in common use, for the sake of a slight saving in machining time.

Aluminum has been greatly improved in the last few years. It is beginning to respond and yield to the ingenuity and long-continued ministrations of the metallurgists. The ideal of combining its lightness with the properties of steel is still far off, but very much of its frailty of character has been removed, without affecting its physical lightness. In the cast state, it is possible to obtain an ultimate tensile stress of 27,000 lb., with an elastic limit of 10,000 lb. and an elongation of 5 to 6 per cent. The elastic limit is sufficiently well marked to indicate that it can be raised by overstrain similarly to that of steel. A new material is thus presented to the engineer.

The peculiar virtue of aluminum for castings is that the weight is not limited by foundry considerations, as in the case of cast iron and cast steel. This renders possible a scientific disposition of material, which is exceedingly important in getting the maximum effectiveness out of a given volume of metal.

It is safe to say that 90 per cent. of engine castings could be made with perfect safety from the same drawings, whether in aluminum or other cast metal, thus reducing the weight by about 60 per cent. By taking advantage of the freedom of design conferred by the strength qualities and light sections permissible in aluminum, this reduction in weight for a given casting

can be considerably improved. In the case of the cylinder, problems of wearing surface arise apart from foundry considerations, but the use of the aluminum cylinder with inserted liners has already proved so successful in aeronautic and automobile practice, as to justify serious attention from the point of view of present and future design. Nor is the high cost of aluminum so serious a bar to its wide adoption as would appear. As against malleable iron, it is about three times more expensive. As it is about one third the weight, castings from the same patterns would cost about the same.

Further, the great facility with which aluminum can be machined and the high costs borne by machine-shop labor are factors in cost of production which often completely offset the higher cost in raw material form. As a practical example arising from aluminum shortage during the war, I was ordered by a British Government department to make certain crankcases and gearboxes of cast iron. A careful investigation showed that there would be no saving in the cost of production and that the necessary extra machine-tool equipment would have entailed a capital expenditure with attendant labor and overhead charges sufficient to make the whole scheme impracticable. The case for aluminum as a structural material rests therefore upon the ease with which it can be worked, the absence of foundry limitations from the point of view of weight in castings such as crankcases and cylinders and the fact that the strength of cast aluminum is now far in excess of what it was a few years ago.

The relative weights per unit tensile strength are as follows, the figures being based upon the ultimate tensile strength in each case:

	Weight per Cubic Foot, lb.	Tensile Strength, lb. per sq. in.	Tensile Strength ÷ Weight per Cubic Foot
Aluminum	170	27,000	159
Gun Metal	500	31,000	62
Malleable Iron	480	40,000	83
Cast Steel	480	60,000	125

As a material for flywheels, aluminum would seem unsuitable, as indeed it is for certain classes of engines, but even for this purpose its inherent strength for a given weight calls for attention. The usefulness of a material for flywheels is determined by the peripheral speed at which it can be run without bursting from centrifugal effects. From this point of view a material is required which has a high ratio of strength to weight per cubic inch. Comparing forged steel with an ultimate strength of 80,000 lb. per sq. in. to cast aluminum with an ultimate strength of 27,000 lb. per sq. in., we obtain the following:

$$\text{Forged Steel } \frac{80000}{0.28} = 2.88$$

$$\text{Cast Aluminum } \frac{27000}{0.1} = 2.7$$

showing that for cases in which space considerations do not set a limit, as in automobile engines, aluminum is far from being a negligible material for flywheels. It is probable that for moderate-sized flywheels, such as are required on direct-driven alternating-current generators, the use of aluminum would be economical in comparison with forged steel. The use of aluminum is also indicated for turbine disks and other cases in which the limit of speed depends upon inherent strength to resist the forces set up by inertia and centrifugal forces. Although these examples are not directly connected with automotive engine design, they are suggestive by reason of the fact that engine sizes and speeds are now rapidly approaching the state in which stresses due to inertia form the basis of engine design.

It is also of interest to compare similarly the strength of forged aluminum with that of various steels, taking for examples a mild carbon steel, a high-tensile carbon steel and a high-tensile chrome steel.

	Weight per Cubic Foot, lb.	Tensile Strength, lb. per sq. in.	Tensile Strength ÷ Weight per Cubic Foot
Aluminum (forged)	170	60,000	350
Mild Steel	480	60,000	124
High-tensile Steel	480	100,000	208
Nickel-chrome Steel	480	135,000	280

The advantages of aluminum in replacing steel forgings are best exemplified by the case of the connecting-rod. In the case of a steel connecting-rod, minimum weight can be obtained only by delicate machining operations, which are further complicated by the necessity for

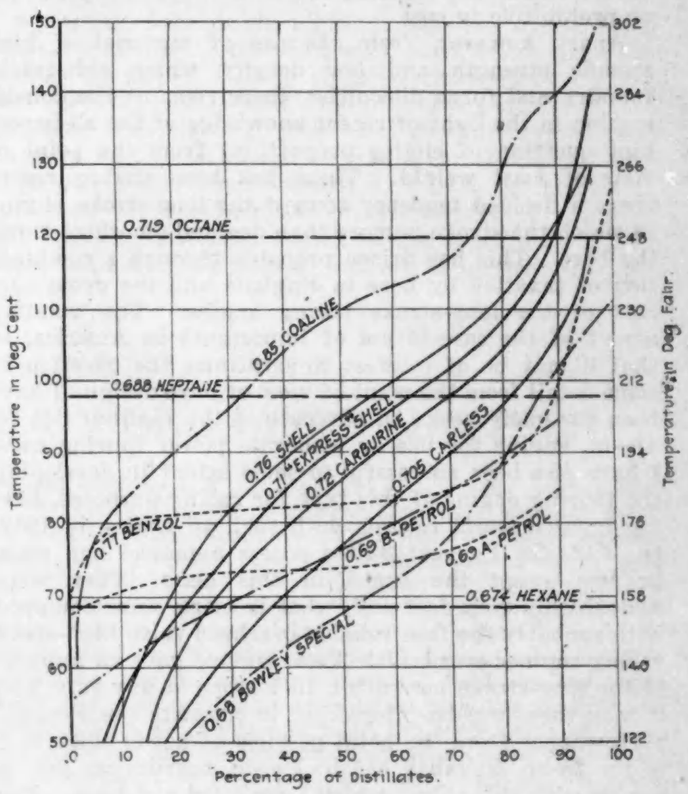

FIG. 14

avoiding sharp radii and fillets to obtain good stress distribution. In other words, the normal steel connecting-rod is about twice as heavy as it would need to be if the material could be properly utilized. With forged aluminum difficulties in respect to machining operations and stress distribution naturally disappear, and the designer has so much "lightness in hand," so to speak, that local high stresses, as for example those in the middle of the big-end cap parallel to the crank axis, can be dealt with liberally without appreciable increase in weight.

In general, compared with steel, the case for forged aluminum rests upon the fact that the strength of members subject to bending or torsion depends upon the ultimate stress and the cube of a linear dimension, while the weight is a function of the square of a linear dimension. Thus, comparing two square beams subjected to bending, one being 1 in. square and the other 1.41 in.

square, the second is twice as heavy, nearly three times as strong and four times as stiff as the first, thus showing the overwhelming effect of dimensions *per se* and the relative unimportance of high tensile strength in members where ample space is available, as in the case of the connecting-rod mentioned above.

Lightness of Construction

The development of the aeronautic engine has focused the attention of designers and the public upon the light engine. In the case of the aeronautic engine, lightness is obtained mainly through machining out low-stressed portions of the various members concerned and because the large size of these engines renders them less susceptible to the limitations imposed upon lightness by foundry and forge considerations. It is not necessary to use twice as much material as is required to make a part which would work perfectly, providing it could be got in place without its being broken by dropping upon the shop floor. In the case of gasoline engines generally, such methods of obtaining lightness are either precluded or prohibitive in cost.

Apart, however, from the use of material of high specific strength and low density which side-tracks foundry and forge difficulties, there remains the consideration in the light of recent knowledge of the all-important question of engine proportions from the point of view of least weight. There has been during recent years a decided tendency toward the long-stroke engine in which the stroke is more than one and one-third times the bore. This has arisen probably through a combination of taxation by bore in England and the great success of the long-stroke racing engine. The taxation aspect of the case is not of importance in America, so that it may be of interest to scrutinize the question in some detail from the point of view of pure design. I have been for many years an advocate of the familiar 3½ by 4¾-in. engine for use on moderate power touring cars. I have also been successful to some extent in developing the touring engine of this type for racing purposes, having created world records with such an engine in 1912. In 1913, the Peugeot 3-litre racers appeared and more or less swept the board in this class. They were undoubtedly very fast and, what is more, were equipped with probably the first reliable overhead-valve high-speed racing engines ever built. Their success gave an impetus to the long-stroke movement, in Europe at any rate, and it may therefore be interesting to compare the Peugeot performance from the point of view of power with that of the 20-hp. Vauxhall 3½ by 4¾-in. touring-car racing engine with side valves which I designed and built. The speed of the Vauxhall on the half-mile track at Brooklands was 100.8 m.p.h., while that of the Peugeot 3⅛ by 5.9 in. of the same capacity with four overhead valves per cylinder was 105. The bodies were as nearly as possible equal in windage and the gear ratios about the same. The difference in speed was due, in my opinion, much more to the overhead valves and consequent better cylinder charging than to any variation in stroke-bore proportions. The difference in power, or more properly mean effective pressure, was probably not more than 10 per cent.

It is obvious that if the mean effective pressure is independent of the stroke-bore ratio, the same power will be developed by engines of the same cylinder capacity at the same speed. Aeronautic engine experience throws considerable light on this question, some of the latest engines being of the short-stroke type and developing mean effective pressures as high as those with longer strokes. In aeronautic engines overhead valves are universally used, so that adequate turbulence is obtained in the combustion chamber, due to the compact shape of the latter. In the case of automobile engines it is undoubtedly easier to obtain turbulence in long-stroke engines with side valves, than in short-stroke engines with side valves, but the difference is only a small percentage and there is, as already indicated, reason to believe that the combustion chamber of a side-valve engine can be modified so as to negative entirely its apparent deficiency in respect to turbulence.

If, then, the problem of stroke-bore ratio can be denuded of its power aspect, which also carries with it thermal efficiency, the ratio of stroke to bore can be settled on the basis of minimum weight and manufacturing convenience. From these points of view the short-stroke engine has everything in its favor. To begin with, the overall length of the engine is usually settled by the summation of the valve diameters, which are necessarily settled by the cylinder capacity, being the same for both long and short-stroke engines of the same capacity. The overall crankshaft length, therefore, is also settled, since the bearing lengths should be proportional to the cylinder capacity, which is independent of the stroke-bore ratio. On the other hand, the larger throw of the crank of the long-stroke engine increases the weight directly, while its inherent extra "crankiness" calls for larger dimensions, if equal stiffness and freedom from vibration are to be assured.

Following the extra crank-throw is the extended section of the crankcase, necessitating extra ribbing and metal for strength and stiffness, extra height on the cylinders due to the longer stroke and longer connecting-rods, if the ratio of crank length to connecting-rod length is to be the same as in the short-stroke engine. The length of valves and camshaft center distance from crankshaft are also increased in the long-stroke engine, the latter calling for considerably heavier timing-gear than otherwise required. The fact that it is far easier to increase the output of a given engine by boring out the cylinders than by increasing the stroke, is in itself an indication of the line of least resistance in design.

I am confident that a restudy of the stroke-bore ratio problem will result in a reaction toward the short-stroke engine, with all its unquestioned charm of sweet running and flexibility. From the point of view of lightness, efficiency and power per weight, it is significant that the latest aeronautic practice should have broken away so completely from the long-stroke superstition by which aeronautic engine design was held in thrall only a few years ago. What is good in principle for aeronautic engines is also good practice in general so far as it can be applied. Above all, the lessons to be learned from the aeronautic engine which are of general application, is that of the proved fact that the short-stroke engine compares favorably with any other type. Because of its inherent lightness and reduction in quantity of material required, it is also economic in the fullest sense, another happy example that the best may be the cheapest.

The short-stroke engine has particular claims to attention from the designers of eight-cylinder car engines. The trouble with such engines is that they involve considerable overall width and can only just be packed into a hood which will conform with the body lines. Even then it is not possible to adopt the high hood which, with suitable treatment, can be made so charming in appearance and to harmonize so well with modern body design. The short-stroke engine permits this desirable end to be realized. It should be remembered that for a given ratio of

connecting-rod to crank length, say 2¼ to 1, a 1-in. increase on the stroke means at least 1⅝ in. on the overall height; ½ in. on the crank-throw and a 1⅛-in. lengthening of the connecting-rod centers.

In the case of an eight-cylinder engine of 3¼-in. bore and 5-in. stroke, with connecting-rods two and one-third times the crank length, the pistons being 1.6 in. from piston-pin center to crown, the distance from crankshaft center to piston crown is as follows:

Crank, in.	2.5	
Connecting-rod, in.	11.5	
Piston-pin to Piston Crown, in.	1.6	
	15.6 in.	Or, say 15.5 in.

With cylinders of 3.75-in. bore and 3.75-in. stroke, or approximately the same capacity, the same connecting-rod to crank ratio and the height from piston-pin to piston crown ⅛ in., we have the distance from crankshaft center to piston crown:

Crank, in.	1.875	
Connecting-rod, in.	8.800	
Piston-pin to Piston Crown, in.	1.800	
	12.475 in.	Or, say 12.5 in.

With the same figures as above but using secondary balancing, allowing a connecting-rod to crank ratio of 1.9, we have:

Crank, in.	1.875	
Connecting-rod, in.	7.125	
Piston-pin to Piston Crown, in.	1.800	
	10.800 in.	Or, say 10¾ in.

The approximate saving in overall width in an eight-cylinder engine, with cylinders at 90 deg., is obviously two and one-half times the saving in height given above, so that the second case would save 4¼ in. overall width over the first, and in the last case the overall width would be reduced by not less than 6¾ in. Designed upon these principles, one of the most serious objections to the eight-cylinder engines goes by the board, and it becomes possible to house it under the same well-shaped hood as now accommodates a four or six-cylinder engine.

Cleanness of design, both outside and in, is being appreciated more every day. The day is rapidly approaching when the public will demand that the automobile shall not be a means of transportation only, but susceptible of artistic treatment in just the same degree as a yacht or a house. The first law of beauty in construction of any kind is that the part shall be the one best fitted to the purpose. In this is postulated a fundamental natural economy, since it implies that excess of material or complication must be avoided.

Some day, perhaps the eight-cylinder car builders will put their heads together and work out a design that will satisfy the æsthetic tastes already inculcated by the six, and embark upon a new and interesting phase of the battle of the cylinders. And when the eight-cylinder car builders have done so and evolved a car that looks thoroughbred rather than cross-bred, it will be interesting to watch the developments of six-cylinder car builders who have been helped as much by the appearance arising from the use of six-cylinder engines as by their intrinsic merits.

S. A. E. ADVISORY COMMITTEE ON ORDNANCE MATTERS

During January the Advisory Committee of the Society on Ordnance matters held a two days' session at the Rock Island Arsenal, all of the members of the committee except one being in attendance. Those present were Herbert W. Alden, the chairman of the Society Committee; George W. Dunham, W. G. Wall, Dent Parrett, C. M. Manly and C. F. Clarkson. The Government representatives with whom the conferences were held were Colonel Jordan, in charge of the Rock Island Arsenal; Colonel Ruggles, chief of the technical staff of the Ordnance Department, and Cols. Moody, Coles, Campbell and Waldman. H. C. McIntyre, formerly a major in the Ordnance Department serving with the American Expeditionary Force and now assistant chief engineer at the Rock Island Arsenal, was also among those present and took a very active part in the proceedings.

Various designs of automotive apparatus are now being perfected by the Ordnance Department for later production. The program involves the development of a hand power cart; ¼ and 15-ton tractors; ¾, 1½ and 3-ton trailer caissons, and 3, 5 and 10-ton high-speed tractors. A large number of engineering problems naturally arise in this connection. For the purpose of giving every possible assistance, the Society committee will hold sessions regularly and give advice in connection with the preliminary specifications, the conduct of tests, the preparation of working drawings and the production of models. Different types of apparatus that have been developed, such as the Mark VIII tank, are now undergoing long-time tests.

Tractor Testing from the User's Standpoint

By Leon W. Chase[1] (*Member*)

Chicago Truck and Tractor Meeting Paper *Illustrated with* Drawings

TWO general plans might be inaugurated in testing tractors; one has for its foundation the design of tractors, the other, their use. Testing tractors to obtain the information desired in their design necessitates testing for friction losses in the bearings, the gears and the entire transmission; for heat losses through the cylinder walls and the pistons, past the rings and through the exhaust gases and for information relative to valve openings, sizes, timings, stems, etc. An unlimited amount of testing can be done on tractor engines to aid the designer. To test tractors and obtain results that will be of value to the user, a test should be inaugurated that determines the reliability, the durability, the power, the economy and the utility of the tractor. It is practically impossible to devise tests which can be standardized and will measure tractor utility and reliability, and it would be a very extensive project to devise tests that would determine its durability. However, tests can very easily be made that will determine the power of a tractor and its engine and the amount of fuel required for doing a unit of work.

University of Nebraska Tests

In devising means for testing tractors in compliance with the State law requiring the University of Nebraska to test one model of each tractor sold in the State, the Board of Engineers has endeavored to devise a measuring unit that is standard for all tractors under all conditions, one that is exact and which eliminates the human equation. These have been divided into two general classes, belt horsepower and drawbar horsepower tests. In addition, there is a miscellaneous test, to be used only in special cases. In preparing the rules and the apparatus for making these tests, the board kept two principal points in mind. One was to devise apparatus and means for eliminating the human element; the other was to devise a standard means of testing each tractor which measured them all with the same unit. In other words, it endeavored to devise a kind of yardstick such that a tractor manufacturer could compare his machine with it so that if the tractor did not measure up to this standard he would believe that it was the fault of the tractor and not of the yardstick.

After considering that the manufacturer might desire to have his engine tested for the power it would deliver at the crankshaft, that it would be impossible to make such a test without removing the entire engine from the tractor because many machines have no means for connecting the crankshaft to any power-driven machine that could be made perfect and that many engines delivered their power to the belt through gears, it was decided that the most just and practical scheme for measuring the horsepower was to take the power delivered by the belt to the brake. This ruling of course complies with the Nebraska law for it measures the power delivered by the engine to the machine the farmer contemplates driving.

After carefully considering the various types of prony, hydraulic and electric brakes, the last was decided upon because of its ease of manipulation, uniformity of operation and accuracy.

Four brake horsepower tests have been adopted for the Nebraska tests. They are, (*a*) a test at the rated load and speed for 2 hr.; (*b*) a test with load varying from a maximum to no load with all adjustments as in the previous test to show fuel consumption under varying loads and also speed control under similar conditions; (*c*) a test under the maximum load with the governor set as in the first test, but during which the carbureter can be adjusted to give its maximum power, and (*d*) a test at one-half load with the governor set as in the first test but with the carbureter adjusted for the most economical operation at this load.

Tractor Operating Conditions

Before describing the apparatus for making the drawbar horsepower tests, a discussion of the field conditions under which the tractor operates will be presented. To those who have not followed tractors while at work, it would seem an easy matter to test the machines when pulling plows. However, the conditions under which plows are used are so variable that no standard measuring unit could be devised; furthermore, it would not be practical to do such extensive field testing work. It does not seem possible to find two fields or, for that matter, a dozen furrows in a single field which will require the same draft and which will offer the same footing for a tractor. For instance, in some tests made by the University of Nebraska in one of the very level and apparently uniform fields of the State and with the same engine and the same plows set at the same depth, the draft was 4.54 lb. per sq. in. in one soil and in an adjoining soil 5.49 lb. per sq. in. of furrow section, a difference of 21 per cent. In another field and with another tractor and another plow, the draft was 8.73 lb. per sq. in. in one soil and in the adjoining soil it was 9.59 lb. per sq. in. of furrow section, a difference of 10 per cent.

These figures only indicate the difference in the draft of the plow and do not in any way show the difference in the energy required to move the tractor over the ground. For instance, in a good, hard stubble field, the tractor moves over the ground with a minimum amount of energy. Under such conditions the tractor should develop an ideal amount of horsepower on the drawbar, while in another field or while in the same field a few days after a rain when the soil is in good plowing condition, the ground would be so soft that the tractor would expend a large proportion of its energy in moving itself across the field. At the same time the ground might be sufficiently soft so that the plows would not require a great amount of energy to draw

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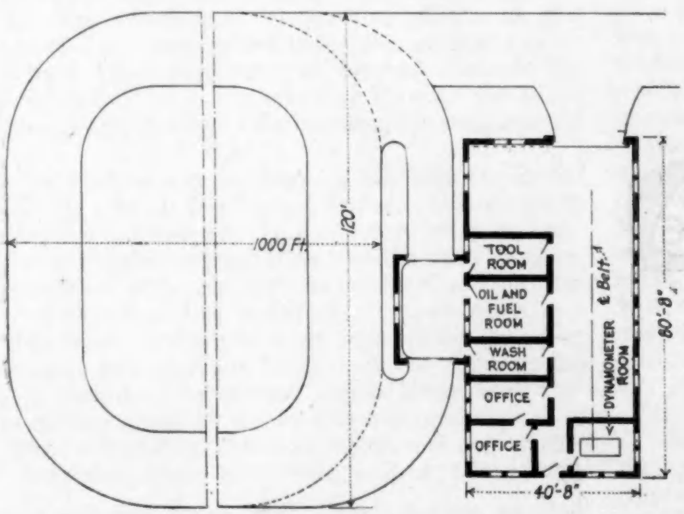
them through the soil. Under such conditions it is apparent that the tractor would be working at a great disadvantage, for a large percentage of the engine power would be devoted to moving the powerplant across the field and not to plowing the soil.

Another consideration disclosing at once the impracticability of testing tractors under field conditions is the fact that, due to climatic conditions, tractors must be tested in the same field having the same soil throughout and by the same operators. This would of course be impossible, for about 265 different tractors are being manufactured in the United States. Should these tractors average 10 acres per day, approximately 2650 acres of land identical in composition, moisture and topography and sufficient apparatus and engineers to make all tests in one day would be required.

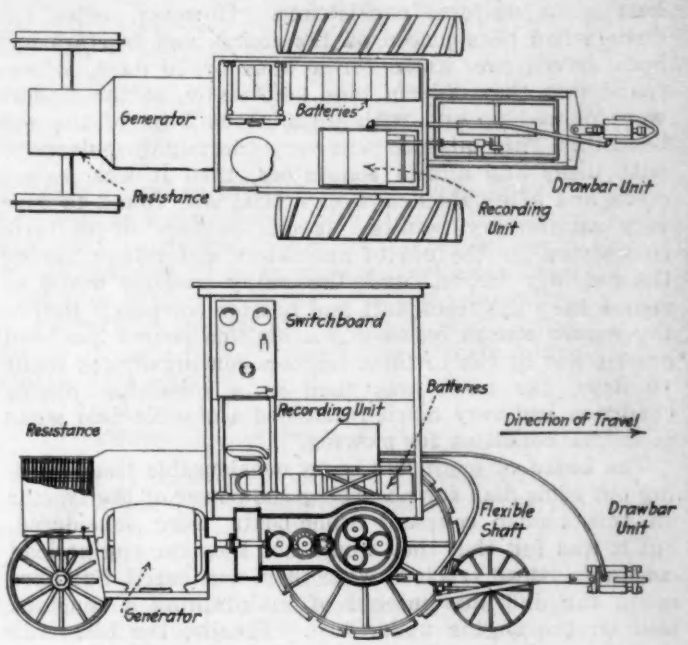
Drawbar Horsepower Tests

Some manufacturers have felt that their tractors should not be tested for drawbar horsepower but for the number of plows or plow-disks they could draw, or by the size of the ensilage cutter, corn sheller or threshing machine they could drive. At first thought, this might be satisfactory from the farmers' standpoint but, because plows have such different drafts and ensilage cutters, cream separators, corn shellers and wood saws, require such a varied amount of power to drive them, no exact comparison could be made. Horsepower is about the only legal unit for measuring power and for this reason it would seem advisable, if possible, to measure the drawbar horsepower of all these engines with the standard unit. To do this the board devised the apparatus described below which has been tried out in several preliminary tests, is now ready for official tractor testing and will be so used as soon as the summer season begins.

The drawbar horsepower tests decided upon by the board are only two in number. One is a 10-hr. test at the rated load of the tractor to be made with the governor set as in the first brake-horsepower test and other tests already mentioned. This will show whether the tractor will or will not continuously carry its rated drawbar load on drawbar work. The other test is to determine the maximum horsepower of the engine. This will consist of a series of short runs with the load increased for each run until the engine is overloaded or the drive wheel slips excessively.



Tractor Testing Course at the University of Nebraska



Plan and Elevation of Tractor Testing Car

It may also be well to give consideration here to other apparatus that was studied while determining the rules for testing. One piece that was given very careful consideration, and for which the Agricultural Engineering Building was planned is that consisting of large drums mounted upon pedestals, upon which the tractor wheels could be placed and held in position. The tractor, while driving through its own wheels, would in turn revolve the drum and brakes which were to be used to absorb this power. To measure the tractive effort a recording scale or dynamometer was to be placed behind the tractor. Such a scheme of testing would measure the efficiency of the gears and the economy and power of the engine. However, it would not indicate the power of the tractor because the plant would not be moving through the field and no energy would be required to drive the front wheels. It would be ideal for the tractor as there would be very little slippage between the tractor wheels and the surface of the drum. It was decided that this device would make an excellent apparatus for laboratory tests but that it would be of no value in measuring the power that the tractor would deliver to the machines which would be used by the farmer.

Another and similar scheme considered was to jack up the rear end of the tractor and bolt drums to the drive wheels which would absorb the power in a manner similar to that of the apparatus previously described. However, this scheme was abandoned for the same reason given in that case.

The Testing Course and Car

It would seem that about the only plan that can be devised for measuring the power delivered to the drawbar of a traction engine in the field is some form of standard field condition. This is practically what was worked out by the board. The field where the tests will be made consists of a ½-mile roadway similar to a race course. However, the surface of this tractor course is composed of about six parts of cinders and two parts of loamy clay. It was at first thought that the cinders alone would make an excellent surface which could be

held at a uniform consistency. However, after the cinders had been placed on the course and tractors had been driven over them for a week or 10 days, it was found that they did not bind sufficiently, so the cinders were plowed up and with them about 2 in. of the soil beneath. This mixture was very thoroughly pulverized with disks and alfalfa renovators, then it was packed down and after about a week's trial was found to give very satisfactory results. It can be kept of uniform consistency by the use of sprinklers and rollers during the hot, dry season, and the rainy seasons would of course keep the track soft and not too compact. Before the winter season began and after this course has been driven over by two or three tractors continually for about 10 days, the track was then in a somewhat pliable condition and very much resembled a stubble field when in normal condition for plowing.

The board of engineers spent considerable time working out some plan for absorbing the power of the tractor on this cinder course. Stone-boats were considered, but it was felt that they would not keep the engine load constant. Dead tractors were then considered, but here again the difficulty entered of maintaining a constant load on the engine under test. Finally, the idea was conceived of removing the engine. the tank and the radiator from the traction engine and inserting in their place an electric generator with the necessary resistance for absorbing the energy it developed, and a design was then worked up.

This piece of apparatus, including the traction dynamometer, is known as the Nebraska dynamometer car. It is composed of an Illinois tractor chassis with a Foote transmission, an electric dynamometer placed in the frame in place of the engine and an electrical resistance substituted for the radiator. Between this chassis and the tractor to be tested a Gulley dynamometer with a rebuilt frame for carrying the driving wheels was placed. Where the engine cab is ordinarily located there is now a place for the operator of the dynamometer car. The controlling switchboard, the recording unit for the Gulley dynamometer and the storage batteries for the generator and the electric lighting outfit complete the equipment. To avoid any danger that this outfit will rear up at the front end the tractor is drawn backward. The direction of energy is thus put into the gears in the same way as was designed for the tractor when driven by its engine.

This apparatus has performed better in all preliminary tests than was thought possible when it was first designed; the only point now in doubt about its perfect success is that it requires a little more power to draw it over the cinder course than was contemplated. Whether or not the car will require too much power for the small tractor, is still a question at this time. However, it is believed that the gears will soon "limber up" and thus the apparatus will draw easier. One interesting element already noticed in the use of this dynamometer is that slippage of the wheels can be detected. This is because the record on the dynamometer recording mechanism shows a line almost perfectly straight when the tractor and the dynamometer car both have smooth wheels and neither unit is slipping. However, when the lugs are placed on either the tractor or the dynamometer car or both the line is very wavy, and when the load on the tractor and on the dynamometer car is sufficiently great to cause slippage, the line is made up wholly of zig-zag marks. Because of this condition the operators of the dynamometer car will have a directly visible indication when either the tractor under test or the dynamometer car itself is slipping.

All of this apparatus can be bought in the open market and this cinder course can be duplicated anywhere in the United States so that this combined plant makes an admirable piece of apparatus for testing tractors. Other plants can be installed and a comparison of the results at the various plants can be readily made.

The University of Nebraska has already spent some $30,000 for this apparatus and for a building in which a large portion of the tractors will be tested. Test work next spring will decide whether or not we have erred in our design. However, it is believed that the apparatus will work out admirably and that the results obtained from these tests will be of value not only to the farming public but to the tractor manufacturers. It is also hoped that the apparatus will be sufficiently practicable and of sufficient value to tractor manufacturers so that they will duplicate and use it at their plants. The University of Nebraska will be very glad to furnish complete drawings and specifications to anyone desiring to make a study of this apparatus and its use.

Composite Fuels

By JOSEPH E. POGUE[1] (*Non-Member*) ANNUAL MEETING PAPER

IN the past year considerable attention in the automotive field has turned to the relation between the internal-combustion engine and its fuel. The rapid rise of automotive transportation in recent years has been the occasion of a country-wide change in the volatility of gasoline, which has attracted widespread interest and raised the problem of better fitting the fuel to the engine, or vice versa, or else striking a compromise between the two.

The problem noted above has been frequently spoken of as one of fuel shortage, actual or prospective. There is no danger of a *material* shortage of automotive fuel in the United States for some years to come, although an *artificial* shortage can readily be created if engine and fuel are permitted to develop at cross-purposes. Expressed in other terms, our liquid fuel resources, lavishly and wastefully drawn upon though they have been, are still sufficient to sustain the needs of the immediate future, if only the products are efficiently utilized. Efficient utilization means coordination between the engine and its fuel, a technical as well as economic adjustment between supply and demand.

As the matter is shaping now, there are three avenues through which this adjustment is tending to come about:

(1) The production of a growing quantity of synthetic gasoline from the heavier oils, through the so-called cracking processes of distillation
(2) Adaptations on the part of the engine to accommodate the efficient utilization of less volatile gasolines and heavier oils
(3) The development of composite fuels or blends, which permit the enlargement and possibly the improvement of the fuel supply, through additions of material not suitable or sufficiently bountiful alone to be of consequence

The future of any one of these three expedients for furthering the advance of automotive transportation depends upon the course of development in respect to the other two, and the final outcome may be expected to be the resultant of numerous factors which cannot be wholly appraised in advance. Not the least of these is the extent to which the whole matter is brought under scientific control by far-sighted and constructive efforts on the part of the fuel and automotive industries acting in common. A final adjustment may be reached through the outworn method of trial and error or through the more modern means employed in meeting tangible engineering issues.

Composite fuels are by no means a new element in the fuel situation, even in the United States. Indeed, much of the gasoline marketed in this country today is composed of straight-refinery gasoline blended with gasoline made in pressure stills, or with casing-head gasoline recovered from natural gas, together with petroleum distillates that were formerly sold as naphtha or kerosene; and even some gasolines are being modified through the addition of benzol. Casing-head blends alone have succeeded in adding about 10 per cent to our total supply of engine fuel. Composite fuels, in which not only distillates of petroleum origin but benzol, coal-tar oils, alcohol and even other chemical products play a part, have of course long been in use in Europe, and some of these met with considerable expansion during the war, especially in Germany. In the past 12 to 18 months, fuel blends containing benzol or alcohol have also come into qualitative, if not quantitative, prominence in the United States, thus drawing attenion to the possibilities of their future importance in this country.

SOURCES OF COMPOSITE FUELS

The resources in sight from which the components of composite fuels may be drawn are mainly three in number:

(1) Crude petroleum, includnig its successor, shale-oil
(2) Bituminous coals, which are capable of yieldnig tar oils, benzol products and other hydrocarbons when subjected to by-product distillation
(3) Organic products rich in sugars, starches or cellulose, especially waste products of organic origin, from which oxygenated hydrocarbons such as alcohols and ethers can be manufactured, chiefly through the aid of bacterial fermentation

As regards the quantity of raw material available, the United States is bountifully endowed in all three respects. Since, however, an extensive and highly organized industrial agency of fabrication and distribution must stand between these resources and the utilization of their fuel potentialities for automotive power, the development of composite fuels becomes dependent not only upon the conditions controlling the growth of the oil-refining industry, the coal-refining industry and a group of activities which may be termed the fermentation industry but also upon the interplay between these three interrelated activities.

The oil-refining industry is the largest, most firmly established and highly developed of the three, and its capacity and industrial ability may be rather briefly dismissed as being sufficiently in mind. This industry turns out four products of major importance to modern civilization, gasoline, kerosene, fuel oil and lubricating oil, not to mention a host of by-products; and its output of gasoline, in consequence, is intimately tied up with the production of joint products demanded by needs scarcely less pressing than automotive transportation. Thus, gasoline has become a product which must be produced, if the market for other oil products is to be supplied; while the oil industry, in addition, has established country-wide machinery for distribution. These economic facts have a direct bearing upon the manner in which composite fuels can be expected to develop; they make it probable that composite fuels, if found desirable, will ultimately be purveyed dominantly by the oil industry rather than by outside activities, under whatever auspices the initial developments take place and without any reference whatsoever to matters of financial control.

The coal-refining industry has thus far been slow of development in the United States. To date it has succeeded in involving only about one-twelfth of the bituminous coal brought into use, approximately eleven-twelfths being still consumed in the raw state; the coke

[1] Industrial economist and engineer, Sinclair Refining Co., Chicago, Ill.

industry and the artificial gas industry are responsible for the advance noted. The production of benzol and related hydrocarbons is mainly dependent upon the progress attained in coal refining.

Up to the present, most of this progress has taken place in the coke industry, where by-product practice is gradually superseding the so-called beehive process, in which benzol and other by-products are not recovered; benzol is now being produced in connection with about half of the coke manufactured in this country. In 1918 the output of benzol from this source was 44,000,000 gal. With by-product practice throughout the coke industry, the output would have been but doubled, scarcely 2 per cent of the quantity of gasoline produced in 1919. The coke industry, therefore, can at best be expected to furnish a quantity of liquid fuel wholly inadequate to have broad significance, except in so far as it may be used as a blending agent to give desirable qualities to other liquid fuels obtainable in larger quantities.

The artificial gas industry was responsible in 1918 for the production of 4,400,000 gal. of benzol, recovered from gas plants operating with by-product recovery. The entire artificial gas industry, however, consumes less than 2 per cent of the country's supply of bituminous coal, and as long as this activity retains its present relatively stationary status the quantity of engine fuel to be expected from this source is practically negligible. There is a possibility, however, that the next few years will see the upgrowth of municipal fuel plants and centralized power stations, operating with by-product recovery, which will give a new source of benzol of greater potential significance than the coke industry in its entirety. Such developments, on the other hand, must of necessity be slow; and should benzol eventually be extracted from the bulk of our bituminous coal, it is evident that, on a basis of 2 to 3 gal. per ton, the supply will even then fall far short of a dominant position as a source of automotive power.

Possibilities of Alcohol

The fermentation industry, notably the branch having to do with the manufacture of industrial alcohol, has been strongly stimulated by war demands, and industrial machinery is now available for the production of considerable alcohol for fuel purposes. The arrival of prohibition has also freed a large equipment from other duties, which might in part be devoted to a similar end. There are serious handicaps of a sentimental nature, however, which tend to bind the manufacture of industrial alcohol with governmental restriction harmful to proper progress; although the war-installed equipment and the cheapness of the requisite raw material may be sufficient to balance these drawbacks.

Alcohol alone can be used to advantage only in engines especially adapted to this fuel, but various mixtures of alcohol, benzol, gasoline or other petroleum distillates and other materials have given promising results. It is of great significance from an economic standpoint that alcohol, benzol and the lighter petroleum distillates such as gasoline and kerosene can readily be rendered miscible. It is probable that alcohol, like benzol, will not come into widespread use as a single fuel, but has a broad significance, for the present at least, only as a blending agent in connection with liquid fuels obtainable in larger quantities.

The quantity of alcohol which will be produced in this country in the immediate future is much more difficult to forecast than in the case of benzol. The United States in 1916, 1917 and 1918, turned out about 50,000,000 gal. of denatured alcohol each year, having jumped from an output of 14,000,000 gal. in 1915 under the stimulus of a demand born of munitions requirements. Much of the industrial alcohol under manufacture today is made from sugar molasses and waste sulphite liquor; while garbage, fruit wastes and ethylene from coal-distillation plants have been suggested as supplementary resources. While the alcohol capacity of the country cannot be closely estimated without a special investigation beyond the scope of this article, the conclusion seems inevitable that for some time to come the available supply of alcohol will bear a close quantitative analogy to benzol, the two combined bulking small when compared with engine-fuel requirements which will approach 5,000,000,000 gal. in 1920.

On the whole, therefore, it may be concluded that benzol and alcohol hold somewhat analogous positions in respect to the supply of engine fuel. Neither can be produced in sufficient quantities in the near future to replace gasoline; both have interesting possibilities in the direction of improving the character of the fuel supply in respect to present engine types. This whole field is undeveloped and stands in need of more research attention than has been accorded it.

Higher Thermal Efficiency of Engines

Composite fuels, while holding out the possibility of improving the adjustment now obtaining between the fuel and the engine, present also the danger of obscuring for a time the necessity of adaptations on the part of the engine in the direction of higher thermal efficiency and lessened dependence upon specialized fuels. Composite fuels, if found to fulfill their initial promise of advantage in utilization, can be developed by the oil industry or, in a more limited manner, by outside agencies; but they can be more readily developed on a large scale by the oil industry, because of its control of working channels of distribution.

In conclusion, while nothing revolutionary may be expected in the way of composite fuels that will displace gasoline, there may come into evidence a steady trend toward a fuel supply of petroleum origin carrying small quantities of other materials, chiefly benzol and alcohol, which will facilitate utilization in the present types of engines. It would be unfortunate, however, if this outcome resulted in a relaxation of the efforts for higher thermal efficiency and for lessened dependence upon specialized fuels, which still remain essential elements in a fundamental solution of the engine-fuel problem.

Springs and Spring Suspensions

By E. Favray[1] (*Member*)

Annual Meeting Paper *Illustrated with* Diagrams

THE chief factors affecting the riding quality of a motor vehicle are: Spring deflection, or amplitude; periodicity, or the number of vibrations[2] per second; and the proportion of the sprung to the unsprung weight. Other factors influencing riding quality are the wheelbase, the tread, the height of the center of gravity of the car and the effect of the front springs on the rear ones. Speed of travel naturally has an effect upon the spring suspension, and therefore on the riding qualities. I will not here consider the effect of speed or speed variation, but shall bear in mind their relative influence at all speeds, with a wheel of a given diameter and the same quality of tire.

The spring having the largest amplitude of deflection for a given load will flex most, for small or large additional upward thrusts of the axle when the wheel encounters obstructions on the road, with the least perceptible disturbance to the car body. For instance, if one spring deflects 1 in. for each additional load of 150 lb. and another moves the same distance for a 200-lb. load, with deflections not exceeding the elastic limit of the material, the 150-lb. or more flexible spring will produce the lesser intensity of upward shock, even though the time interval during which the upward thrust acts is longer.

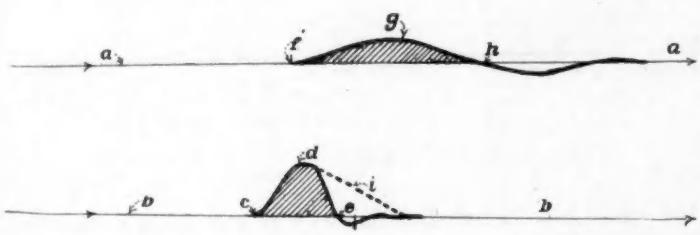


Fig. 1

Vibrations Per Minute

The periodicity, as well as the amplitude of the deflection, depends upon not only the length, thickness and width of the spring-leaf, considering a single-leaf spring for the present, but also the load that it supports. The greater the static load and the greater the static deflection of the spring, or the position of rest when the load is applied, the slower the vibrations per minute. Increase the static load and the deflection will increase the fiber stress in the spring; so we can state that the greater the fiber stress the slower the periodicity.

It is not so much the height through which the body is raised as the rate of speed at which it is raised that affects the passenger's comfort, although both are of great importance. The final requirement of a good spring suspension is that it impart to the car body the least amount of upward motion and at the slowest speed. Tests[3] have disclosed the fact that the axle, when the wheel travels over an obstruction, causes the wheel to jump over the obstacle in each case and come to the ground beyond it (See Fig. 1); and it appears that the shape of the obstacle makes no appreciable difference, unless its slope is very gentle. It has also been shown that the wheel reaches the top of its upward motion almost before the car body begins to move upward, and that when the wheel has completed its "jump" and returned to the ground the body has traveled only 40 per cent of its upward path.

In Fig. 1 the lower line denotes the path which the center of the wheel describes when surmounting the obstruction, while the upper curve shows the path pursued by a point of the frame or body just above the center of the wheel. Tests have also shown that when a stiffer spring is used the oscillations of axle and body cease sooner, but that the acceleration of the upward motion of the body is greater, which means more discomfort.

Relation of Sprung to Unsprung Weight

Fig. 2 illustrates graphically the importance of keeping the proportion of sprung to unsprung weight very great. A comparatively heavy weight is represented by w, while w_1 is a small weight. In the first instances a small weight at the bottom, corresponding here to the axle and the wheel, will have relatively little effect upon the greater weight above, since the latter is so much heavier; the greater weight will be influenced relatively little by what the lighter weight may do. In the second case we have a heavy weight at the bottom, which, if it begins to move upward, will continue to do so, irrespective of the small weight above, and no matter how weak or strong the spring above may be, it will carry the small weight along with it. Therefore, as far as riding quality is concerned, a lighter unsprung weight will be superior, since it has the least effect upon the sprung weight.

Let us consider the following example, on each rear wheel:

Weight of car, lb.	4,000
Weight on each rear wheel, lb.	1,200
Unsprung weight for each rear wheel, lb	300
Sprung weight for each rear wheel, lb.	900
Horizontal speed of car, ft. per sec.	72
Approximate speed of car, m.p.h.	49

In considering the action of a wheel of a moving car when striking an obstruction over which it rises, so many

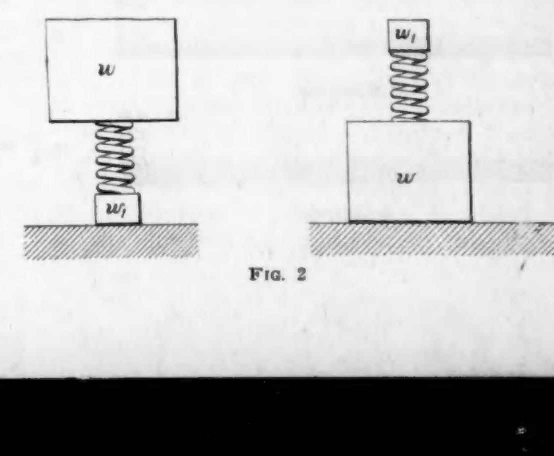


Fig. 2

[1] Consulting engineer, New York City.

[2] The term "vibrations" is used instead of "oscillations", since the best writers in physics use "oscillation" to mean one-half of a complete vibration; thus, a vibration means two oscillations, one up and one down.

[3] *Transactions* of the Institution of Automobile Engineers, Vol. VII, page 451.

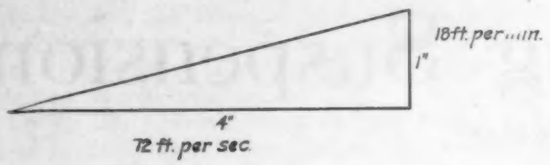


FIG. 3

modifying factors are involved that it is doubtful whether, with the data available at present, a correct and complete mathematical analysis is possible. Still, certain assumptions can be made that more or less approximate true conditions, and a result secured such as will give some idea of the magnitude of the forces involved. When the wheel, moving along a horizontal surface, strikes an obstruction, there is a tendency to reduce the speed of the car because there is imparted to the wheel of the car potential energy that has been acquired at the expense of the kinetic energy of the car. Since the mass of the car is large compared with the unsprung weight per wheel, we can assume as a first approximation, that the car speed remains unchanged while passing over the obstruction. The simplest assumption possible is that the path of the hub of the wheel is a straight line inclined at a definite angle to the horizontal. This is equivalent to assuming that the wheel is traveling up an inclined plane.

When the wheel strikes an inclined plane there is a tendency to retard its motion, since it cannot acquire upward velocity in zero time. This brings about a changed deformation of the tire and a slight distortion in the supports dragging the axle and the wheel for an instant has a velocity which is slightly less than that of the car. As the distortion in the dragging supports disappears, due to the resiliency of the material, the wheel again catches up in speed and finally when the distortion has completely disappeared the speed of the car and of the wheel is identical and the latter has attained its maximum vertical velocity. From the foregoing it follows that the vertical acceleration is variable.

Let us further assume that the wheel in passing over an obstruction rises through a vertical distance of 1 in., and that in doing so it travels a horizontal distance of 4 in. (See Fig. 3), and that at this instant the wheel has attained its full vertical speed. The wheel must then have a vertical velocity of 18 ft. per sec. Since this vertical velocity was acquired while the wheel traveled a horizontal distance of 4 in. or 1/3 ft., the time

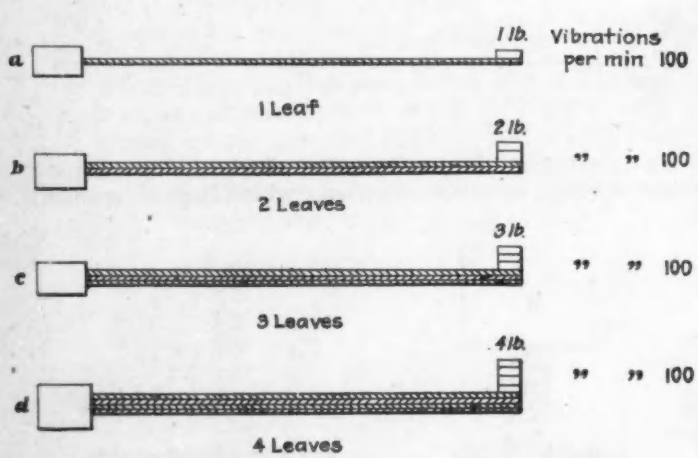


FIG. 4

interval is 1/216 sec. Then the average acceleration is:

$$a = \frac{v}{t} = \frac{18}{1/216} = 3888 \text{ ft. per sec. per sec.}$$

Because the acceleration varies from zero to zero, the actual acceleration for some points of the path must be considerably greater than the value found for the average. Ordinarily the axle will not rise 1 in. in traveling 4 in. horizontally, but this rather unusual case was purposely selected to indicate roughly the magnitude of accelerations that do occur. The upward acceleration of the axle amounting to anywhere near the above figure, will exert a tremendous force against the body and tend to give it a definite upward acceleration and velocity. This upward force of the axle can be reduced by decreasing the unsprung weight and also by some means that permit the axle to move slightly backward with respect to the frame, when subjected to an upward thrust. This is often accomplished in practice by raising the front end of the spring, for instance.

One function of the spring is to decrease the upward velocity and acceleration of the sprung weight, or car body, and to increase the distance of its horizontal travel while in a raised position. The spring thus transforms the vertical distance of the body deflection, either up or down, into a horizontal distance, whenever the wheel surmounts an obstacle or sinks into a rut. In Fig. 1, if

FIG. 5

the wheel hits the obstruction at *c* it will quickly rise to *d*, that is to say, in a very short horizontal distance, and will soon thereafter drop back to its normal path *b b*. The body above will start to rise at *f*, reach its highest peak at *g* and then slowly descend to the normal path, *aa*. It is evident that the later the curve of the body-path begins to rise, after the wheel starts its upward acceleration, the lower the peak *g*, or the longer the time required for it to reach its maximum height. The greater the horizontal distance *f h* the better will be the riding quality of the car. If no kinetic energy is absorbed by the spring the area of the shaded section under the lower curve will be approximately equal to that under the upper curve, but the greater the amplitude of the spring deflection and the lower its periodicity, the lower and longer will be the upper curve, described by the body, and the higher and longer the lower curve described by the wheel center. A flexible spring, one with a large amplitude, will permit the axle to fly up very rapidly and to a great vertical height; while a low periodicity will return it to the normal path more slowly, as shown by dotted line *di* (Fig. 1). Even if the upper curve were higher, if it is proportionately longer before returning to normal, a more favorable riding quality will result.

While the formula for finding the periodicity of a spring, or number of vibrations per minute, can be found in any textbook on the subject, let us analyze it by simple practical experiments. If we take four leaves of spring steel, all of which are identical as to length and thickness, and if we hold one end fixed and leave the other free to vibrate, we find that without any load the periodicity will be the same whether we have one leaf or two, three or four leaves together. If the leaves

are straight and smooth, so that there is no friction between them, we will also find that the amplitude of deflection is practically the same for the same load per spring, as is also the number of vibrations, before the spring finally stops. If we now place a weight of 1 lb. on the free end (See Fig. 4), we find the amplitude of deflection larger but its periodicity lower. For instance, in one case, without any load on the free end, a certain spring 30 in. long had a periodicity of 126 vibrations per min. regardless of the number of leaves. When 1 lb. was placed on the free end, the vibrations were reduced to 100; with 2 lb. to 86; with 3 lb. to 78; with 4 lb. to 73, etc. When 1 lb. was placed on the spring composed of two leaves, the periodicity was 112, but with 2 lb. it was 100; with three leaves, 1 lb. reduced the vibrations to only 116; 2 lb. to 108; 3 lb. to 100, etc. This is shown graphically in Fig. 4. With 1 lb. on one leaf, 2 lb. on two leaves, etc., the periodicity as well as the amplitude remains the same[4] By shortening the leaf to 24 in. the vibrations were increased to 196, without load, while with a 1-lb. load they dropped to 142; with 2 lb. to 120, and with 3 lb. to 110. Two leaves shortened to 24 in. gave a periodicity of 172,

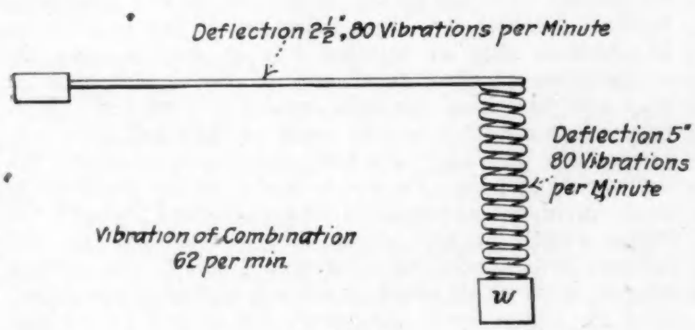


Fig. 6

with a 1-lb. load; 142 with 2 lb., etc. Three leaves gave a periodicity of 176 with a 1-lb. load; 142 with 3 lb., etc. We can state, therefore, that, everything else being equal, a heavier load, which is always accompanied by a larger static deflection, causes a greater fiber stress in the spring; and the heavier the load, or the greater the deflection or the greater the fiber stress, the slower the periodicity.

Let us now consider the effect of friction between the leaves. In the simple experiments covered by *d*, Fig. 4, rubber bands were tied around the springs to create friction between the leaves. We still obtain vibrations at the rate of practically 100 per min., but instead of being able to count the vibrations for a whole minute, they are damped out in about 10 sec. and the static deflection is less than it was without the rubber bands. It is thus evident that friction between the leaves reduces the amplitude of deflection and the number of spring-inertia vibrations before the spring comes to rest.

[4] It might be mentioned that these tests were not carried out with extreme accuracy; also, the load was not pound weights, but blocks of uniform size and small weight.

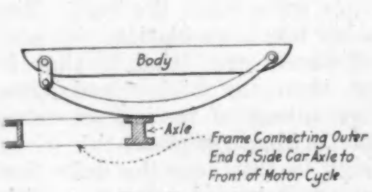


Fig. 7

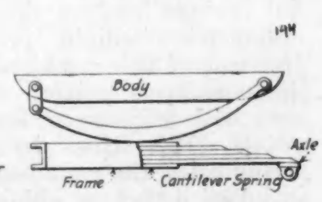


Fig. 8

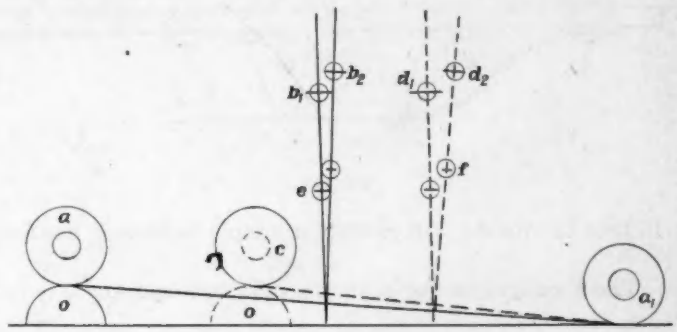


Fig. 9

Spring Inertia

Another simple experiment showed that a thin spring, such as is shown in Fig. 5, when hit quickly on the free end, first assumes the shape *a*, then *b*, and finally *c*; when it will continue its vibrations as in *c*, gradually declining in amplitude until it comes to rest. In other words, when quickly applying the load at the free end, the spring will not at first be flexed over its entire length but gradually flex from the point where the load is applied toward the fixed end.

If the spring is very thin, air resistance may be the chief cause of this retarded deflection of a portion of the spring, but inertia is an important factor. We may ask whether something else does not prevent instantaneous flexure of the entire length. It seems that scientists are agreed that no delay is caused by molecular intercommunication in the spring material itself. On the other hand, Thomson and Joule discovered that metal, when subjected to stress within its elastic limit, experiences a very small change in temperature, diminution for tension stresses and an increase for compression stresses, and that the change in temperature is proportional to the change in the stress applied. I believe it was Lord Kelvin who found a difference in the electrical resistance of stressed and of unstressed metal. It would be interesting to hear from members who have conducted experiments, whether to their knowledge springs were ever set vibrating in the laboratory at such a rate as to attain an acceleration of 3000 or 4000 ft. per sec. per sec. at their free ends. The fact that a spring does not respond instantaneously over its entire length, as disclosed from the experiment mentioned before (See Fig. 5), would indicate that if we set the spring vibrating at a slower rate it would be more satisfactory.

Fundamentals of Periodicity

We may briefly analyze the fundamental equation governing the time t in seconds, required for one period or one oscillation down and one up. Assume a helical spring suspended vertically and supporting a mass m whose weight is mg lb.; then the mass will come to a position of equilibrium when the restoring force of the spring is equal to mg. According to Hooke's law, if f is the force in pounds required to elongate the spring one unit, then fs will be the force required to elongate the spring s units. Since the spring obeys Hooke's law, the restoring force tending to bring the mass to its position of equilibrium is proportional to the displacement from the position of equilibrium, and, therefore,

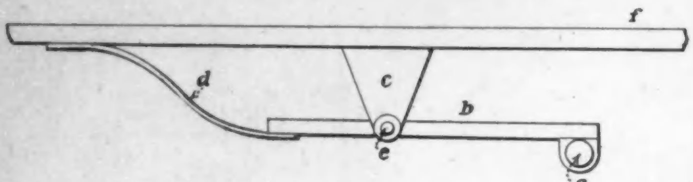


FIG. 10

if free to vibrate will execute a simple harmonic motion.

The acceleration for a simple harmonic motion is $\frac{4\pi^2}{t^2}s$.

Therefore, the force for the displacement s is $m\frac{4\pi^2}{t^2}s$, but this must be equal to the restoring force. Therefore

$$m\frac{4\pi^2}{t^2}s = fs, \text{ from which } t^2 = \frac{m4\pi^2}{f} \text{ and } t = 2\pi\sqrt{\frac{m}{f}}$$

The mass and the restoring force alone govern the periodicity. The restoring force in a leaf spring depends on the length, thickness and width of the leaf and on the modulus of elasticity.

In comparing leaf and coil springs proper allowance must be made for the spring mass involved, but in general the formula governing all vibrating bodies is:

$$t = 2\pi\sqrt{\frac{\text{Resistance}}{\text{Restoring force.}}}$$

In this connection, it will be interesting to compare the periodicity of a leaf spring having a certain deflection with that of a coil spring with the same deflection. Suppose a single-leaf spring, like those shown in Fig. 4, be given by the application of a certain weight a deflection at the end of the spring of 2.5 in.; and suppose that upon setting it vibrating its periodicity is found to be 80 vibrations per min. By taking a helical spring of such length and stiffness that the same weight deflects or elongates it also 2.5 in., we obtained by actual test a periodicity of 120. Of course, this amount would vary somewhat in changing the weight of the spring, but it can be stated that ordinary coil springs used in practice, for supporting and cushioning similar loads, have a very much larger number of vibrations per minute than leaf springs carrying the same load. To reduce the number of vibrations to that of the leaf spring, 80 per min., it was necessary to increase the length of the coil spring to such an extent that its static deflection for the same weight amounted to about 5 in., or double the deflection of the leaf spring.

If the two springs, having the same periodicity, are now taken and connected together in series as shown in Fig. 6, and the same weight be suspended therefrom, we obtain a combined periodicity[5] of 62. It is thus seen that a combination of two springs reduces the periodicity. It has been stated by some engineers that the use of coil springs at the ends of the leaf spring, such as are employed in certain coil-spring "shock absorbers," does not reduce the periodicity of the combination. I have found in every case that such application brings about a diminution in the number of vibrations per minute, although in some cases, when the proportion of coil-spring deflection to leaf deflection is very small, as with the coil-spring "shock absorbers," the decrease is almost negligible. It has been found that by attaching a coil spring at the end of a leaf spring, especially when the coil-spring deflection or elongation per unit weight is considerably greater than that of the leaf spring, the coil spring does reduce the acceleration of the free end of the leaf spring. When several leaves are used and considerable frictional surface between the leaves is thus introduced, the coil spring will absorb slight vibrations without transmitting them perceptibly to the leaf spring.

[5] The experiments were crude, hence an error of 2 or 3 per cent is likely.

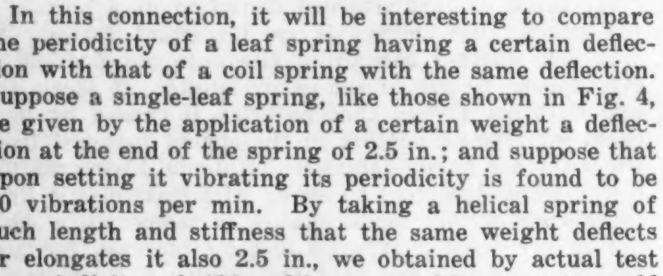


FIG. 11

This is precisely what happens in practice by the application of coil springs to the ends of leaf springs. They absorb the small rapid vibrations, while the leaf spring is practically at rest, but in the case of large vibrations, the leaf-spring deflection is the governing factor. Nevertheless, the deflection of the coil spring is added to that of the leaf spring, and reduces the acceleration of the "free" end of the leaf spring or that end to which the coil spring is attached. It is believed that better results could be obtained with supplementary springs attached somewhere between the leaf spring and the axle, for in that case the small vibrations would not of necessity be transmitted through the entire length of the leaves before reaching the coil spring; and in addition, this would add to the sprung weight as far as its effect on the coil spring is concerned, since the latter would then carry the weight of the leaf spring.

Having found that the ordinary coil spring of a given deflection has a higher periodicity than a leaf spring, it appears that by substituting a separate leaf spring between the main spring and the axle, better results might be expected. My experience in this connection may be of interest. While engaged recently in redesigning a rather heavy motorcycle with a side-car which, among other defects, rode most uncomfortably, I found that the side-car frame was rigidly attached to the axle (See Fig. 7). There were leaf springs 38 in. long between the frame and the side-car body, but the proportion of unsprung to sprung weight was comparatively large, because the frame rested directly on the axle. There was available also an experimental side-car frame terminating in a short, stiff quarter-cantilever spring attached to the axle. The body was mounted directly on the frame. In testing this on the road it was found to be as bad as an empty truck as far as riding quality is concerned. The body was consequently removed and the first side-car body with semi-elliptic springs placed on the frame, which terminated in the cantilever spring, as shown in Fig. 8. This gave a combination of short flat springs between the axle and the frame with the ordinary semi-elliptic springs underneath the body. The first test of this combination was a revelation. It was the unanimous opinion of those who tried it that it rode with greater comfort than any high-priced automobile. This shows the advantage of having an extra spring below the main one. In the first place, this gives an added deflection, although in this case the deflection per unit weight was very small indeed compared with

the main spring, and this reduced the periodicity of the body, and secondly the acceleration and velocity of the free ends of the main spring were therefore less.

Thus, to obtain increased comfort, one suggested method is to provide an auxiliary spring, preferably of the leaf type. Another great advantage of the leaf spring is that the inter-leaf friction reduces the vibrations to a very small number before the body comes to rest, while with coil springs the vibrations continue for a long time, and this is to be avoided in car suspension. On the other hand, excessive friction between the leaves interferes considerably with spring deflection and renders the spring much stiffer, thus imparting greater shocks to the sprung weight. It seems that comparatively little friction between the leaves, as for instance when they are lubricated, is sufficient to reduce the total number of vibrations before coming to rest to that required for comfort, except when going at high speed over bad roads, in which case added friction is advantageous. Friction in the spring shackles also influences the deflection; likewise the angle of the shackle with respect to the spring. If this angle is such that the stress applied to the spring is at right angles to the leaves, it will impart purely flexure stresses to them. If at a greater angle, it will cause, in addition, buckling stresses; and if the angle is smaller, tensile stresses.

Another point worthy of notice is that, with leaf springs composed of a number of leaves, a certain load is required before the spring begins to deflect, due to the inter-leaf friction. This is a serious drawback in both automobile and truck applications; in the first case it renders a touring car less comfortable unless fully loaded, and in a truck it means excessive jars and shocks when empty or only partially loaded. The latter, under such conditions, might as well have no springs, in many cases, since at such loads there is very little, if any, spring deflection. The result is rapid deterioration.

The effect of the height of the center of gravity and of the length of wheelbase is illustrated in Fig. 9. Consider first a and a_1, which represent respectively the front and rear wheel of a car having a long wheelbase. Let b_1 represent a point in the body of the car. When the front wheel surmounts an obstruction o, point b_1 will be moved to b_2, due to the angular rise of the front wheel with respect to the normal position of the rear wheel. In this case the backward distance is very small compared with that shown in d_1 and d_2, which represents an analogous case with a short-wheelbase car. It is the backward or transverse motion that causes the passenger discomfort, rather than the vertical motion, everything else being equal, and naturally the greater the horizontal distance through which the passenger is moved, the greater the discomfort. These remarks apply also to the width of tread, as affecting side-to-side motion. The illustration moreover discloses what would happen if the points b_1 and d_1 were lowered, or the center of gravity of a car were lowered; the horizontal or transverse motion is reduced when the car wheels rise over an obstruction or sink into a rut.

Possible Future Constructions

Having so far considered the inherent qualities of the present ordinary types of suspension and seen how the addition of auxiliary springs will improve the suspension, we will now consider possible future constructions where no auxiliary springs are required, keeping in mind that springs are not sensitive at light loads, do not respond instantaneously under very rapid applica-

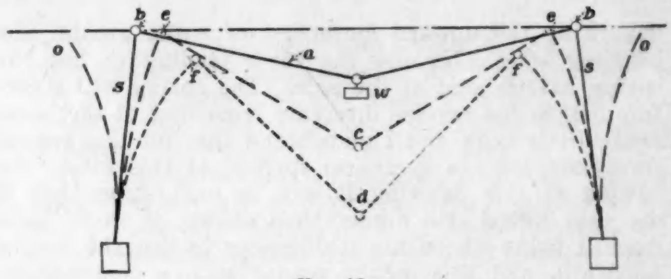


Fig. 12

tion of load or less energy is required to deflect a spring mass at a slower rate of speed and that a relatively great length of spring is necessary to obtain a large deflection and low periodicity. The spring deflection increases uniformly with the load; hence, to obtain more comfort, a greater total deflection must be provided. This is objectionable for several reasons, such as added cost and weight and increased inertia.

Let us first consider the effect of a lever arrangement, such as to permit a large movement of the axle compared to the frame and result in small spring deflection (See Fig. 10), to ascertain whether this holds any promise for spring suspensions of the future. Here a represents the axle, b a lever or solid beam pivoted to e, c a bracket supporting the lever and attached at its upper end to the frame f, and d a spring of any description. By moving the axis e farther away from the axle, the spring flexure can be made very small, if desired, for a large upward motion of the axle. But the one great disadvantage of such a system is the inertia of the beam b. A sudden blow of a certain rapidity and intensity will break the beam at its pivotal point, scarcely flexing the spring. For motor vehicles, therefore, where the upward acceleration of the axle is high, it is believed such a construction is impractical and needs no further comment. However, if the beam b were an ordinary cantilever spring with its end pressing against another spring d, which, if desired may be a coil spring, very much better results could be obtained.

Next, consider a semi-elliptic spring a, attached to the frame (See Fig. 11), with suspension members c connecting the spring-eyes to the axle d. This system offers at first sight the advantage of being very sensitive at light loads, when the axle is only slightly higher than the spring-eyes. In this case, when the axle is moved upward to e, the spring-eyes will move from h to i; this is a smaller distance than the axle travels and the greater the distance between the spring-eyes the smaller the travel of the latter for a given distance of axle movement, based on the principle of the catenary. Turnbuckles t can be provided to tension the spring in the amount required for a satisfactory static deflection. The disadvantage of this construction is that the pull or tension exerted on the spring leaves is not conducive to purely flexure stresses, since the greater the spring

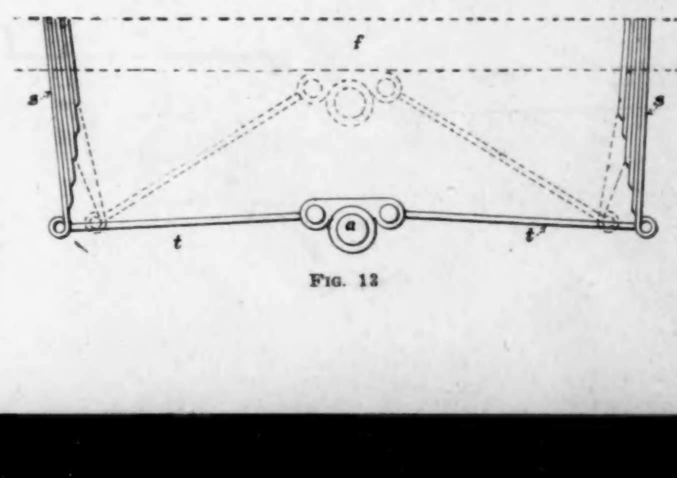


Fig. 13

flexure or the upward movement of *e* the greater the buckling stress, because the force tending to flex the spring has its seat at the axle. The spring will therefore flex in the reverse direction from that of the usual semi-elliptic type, and I understand that for this reason spring-makers are averse to springs of this kind. By looking at this drawing it will be understood that if the axle moved still higher than shown, it would soon reach a point where the tendency is to flex the spring backward, and this would render such a construction impractical. A different arrangement which in my opinion offers a field for future development is that disclosed in Fig. 12. Suppose we take springs similar to those used in the former experiments and shown in Fig. 4, and attach them rigidly to the frame in a vertical or nearly vertical position, and then connect the free ends of such springs by a cable *a*, or two rods pivoted in the center. If we now apply a load *w* at the center of the cable, we shall obtain a certain deflection depending upon the magnitude of the load, the span between the spring ends *bb* and the tension in the springs *ss*. Suppose we now increase the load so that the center of the cable will be deflected to *c*; the spring ends will move inward from *bb* to *ee*, a distance small in comparison with the amount *w* has moved. Thus, for a large deflection in the center we have a small spring flexure. If now the load is still further increased, the weight will reach *d*, while spring ends *e* will move to *f*. It is noteworthy that while the distances *w* to *c*, and *c* to *d*, are equal, the spring deflection is about double in the second instance. In a rough test under these conditions the deflections under various loads and their periodicity were measured, the distance between the center of cable and spring-eyes being 2 ft. The spring tension was adjusted so that the same weights as were used in previous experiments (See Fig. 4) gave deflections measured vertically at the center of the suspension where the load was applied and periodicities as given in the accompanying table:

Number of Weights	Deflections, in.	Vibrations per min.
1	3.50	96
2	5.75	82
3	7.50	76
4	9.00	73
5	10.10	70
6	10.80	69

When the distance between the spring-eyes and the point of load application was increased to 30 in., the periodicity was found to be a trifle lower when the tension was adjusted so that the application of the same load gave a similar load-deflection. With a greater distance, a greater spring-tension was necessary for the same load-deflection. However, with the same spring-tension maintained, the deflection under a given load will be greater.

Fig. 13 shows one way in which a type of suspension can be applied to a motor vehicle. In this case *f* is the frame, *ss* the springs, *a* the axle, and *tt* the tension bars. In this example the springs, as seen, deflect only about one-third the distance of the load deflection. The construction illustrated can be simplified by using radius-rods, commonly installed on motor vehicles, to take the place of one spring and one tension bar, as shown in Fig. 14. Fig. 15 shows a similar arrangement for the front spring; *f* is the frame, *s* the spring, *t* the tension rod, *r* the radius-rod and *a* the axle. In Fig. 14 the arc xx_1 shows the path of the axle travel when the spring is flexed. To make the axle go backward as well as upward when meeting an obstacle, the anchorage of the radius-rod can be moved higher, to *z*, for instance, in which case the axle will travel along

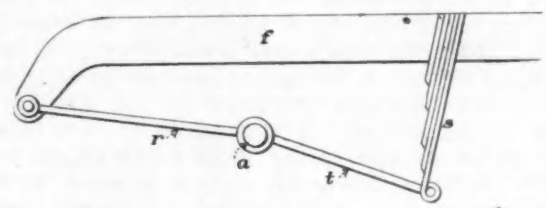


FIG. 15

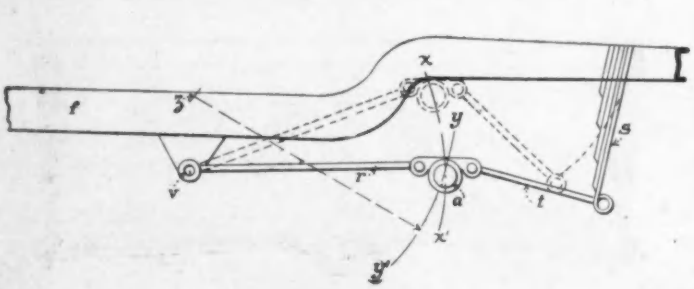


FIG. 14

the arc yy_1. In addition to overcoming the defects of the ordinary spring suspensions before mentioned, with this type of suspension the weight of the spring is removed from the axle and added to the sprung weight, which we have seen before is conducive to added comfort.

Let us now examine the tension in the rods and the bending moment of the spring at the spring seat. We will assume that the pull exerted is at right angles to the spring leaf, as it can be in practice with this type of construction (See Fig. 13). Suppose the weight *w* at normal load is 1000 lb. and at maximum deflection 2000 lb.; the length *l* of the tension rods, that is to say, the length between the axle and the spring-eyes, is 24 in. If the maximum deflection is 10 in. from the horizontal position (See Fig. 16), the tension *t* in the rods pulling against the springs can be found by direct proportion:

$$\frac{l}{t} = \frac{2d}{w} \text{ or } t = \frac{lw}{2d} = \frac{24 \times 2000}{2 \times 10} = 2353 \text{ lb.}$$

If the free length of the spring, from the spring-eye to the seat, is 14 in., or approximately 1.17 ft., the bending moment at the spring seat will be 2353 x 1.17 = 2750 ft.-lb. approximately. This would compare with a semi-elliptic spring somewhat as follows: Length of spring approximately 56 in., with a free length on each side of 26 in. Spring deflection 1 in. for every 200 lb. of load. Normal 1000-lb. load deflection 5 in. and maximum total deflection 10 in. at 2000 lb. We have at the maximum deflection 1000 lb. at each side of a spring length of 26 in., or 2.17 ft., and the bending moment is 1000 x 2.17 = 2170 ft.-lb., about one-quarter less than with the above suspension. In spite of that, the weight of the two springs of Fig. 13 can be made less, since their combined length is not much more than one-half that of the semi-elliptic. The weight of the constructions shown in Figs. 14 and 15 is approximately one-half that of the suspension of Fig. 13.

With these types of suspension provision must be made for the lateral strength of the frame, but there

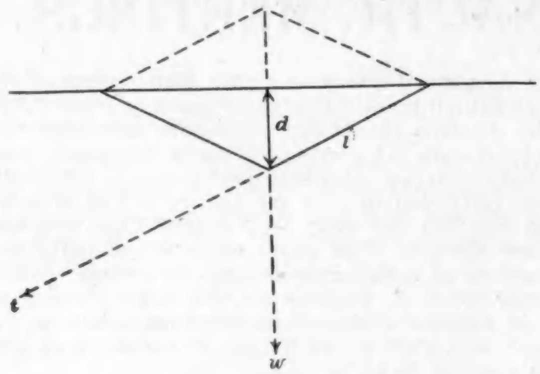


FIG. 16

are various simple means available for accomplishing this. One feature which I felt uncertain about without a practical model was the amount of rebound as compared with the ordinary semi-elliptic spring. I accordingly applied this suspension to the rear of a Ford car and was surprised at the absence of large rebounds. Evidently the friction between the leaves was sufficient to dampen the oscillations very quickly.

To summarize, the advantages that the suggested suspension appears to offer are: Sensitiveness at light loads; a minimum flexure for a maximum of axle motion; sensitiveness at all loads against quick upward thrusts of the axle; reduced weight of the unsprung mass; the spring deflection does not increase uniformly with the load but provides larger deflections at light loads and proportionately small deflections near the maximum load, and a saving in spring weight. The practical development of this type of suspension will no doubt disclose a number of minor problems, but none of these is in my opinion impossible of solution. The type appears to me to be one most promising as yielding improvements over existing suspensions.

LONDON-PARIS AIRPLANE PASSENGER SERVICE

AIRPLANING from London to Paris and vice versa is becoming quite a matter of course. The Hounslow aerodrome was one of the first and perhaps the most famous training ground in England for military aviators at the beginning of the war. Now it has become the first real official aerial link for commercial and passenger service between London and Paris. Hangars and huts that were built for use in the early months of 1914-15 still circle the huge flat meadow that is about 10 miles from Piccadilly Circus, in the heart of the metropolis.

Since Aug. 27 last, punctually at 12:30 every day, no matter how stormy or how misty the weather, an air 'bus has taxied off on its non-stop flight to Paris, carrying its complement of passengers and light luggage. Only once since the starting of the service has a pilot been compelled to make a forced landing, and then the delay was so brief and the rectification of the engine trouble so slight that scarcely any inconvenience was caused.

As yet the station at Hounslow has no elaborate buildings or waiting rooms. The fleet of flying "buses" is sheltered beneath a huge shed that has been used for war purposes almost 5 years.

The machines that fly to Paris every day are of two sizes, one holding the pilot and two passengers, the other affording comfortable space for four or six passengers. The fare charged is $100 per person, and the flight is accomplished in less than 3½ hr. from Hounslow to Le Bourget, a suburb of Paris.

The enclosed carriage of these passenger machines is extremely comfortable, furnished with deep barrel seats upholstered in leather, and with a writing table in front, upon which is a speedometer. As yet, only a small amount of luggage can be carried, passengers being limited to about 20 lb. each.

The route taken from Hounslow across England to the Channel is over Kenley and Lympne, near Folkestone, to Marquise, near Boulogne and thence to Paris.—*Evening Sun* (New York).

MOTOR BOAT AND AERONAUTIC MEETINGS

THIS year the Society returns to its former practice of holding a meeting in connection with the Motor Boat Show at New York City. On the evening of Feb. 25 a meeting devoted to consideration of motor boat problems will be held at the Engineering Societies Building. The program includes talks and films contributing to the fund of information on marine engine testing, designing and motor boat performance. Much of this knowledge which formerly was not available is now in shape for presentation at a gathering like this and for publication in THE JOURNAL. With the capture of heavy-oil engines of German design, engineering practice and performance along these lines has become better known. Many suggestions are raised concerning the effect, if any, which aircraft engine design and practice have had upon speed-boat engine design. These and many other timely subjects will be brought out by the papers presented at the Motor Boat Meeting. The members are expected to take advantage of the opportunity for discussion and to bring out interesting data on motor boats and their power equipment. The program has been arranged to point the way by providing a forum where motor boat problems for the ensuing year can be discussed.

The aeronautic meeting on March 10 at New York City will consist of an afternoon technical session in the Engineering Societies Building, followed by a dinner and reception at the Hotel Astor at 6:30. Contrasted with popular opinion, the men engaged in the aeronautic industry maintain that aircraft will play as important a part in civil requirements as it has taken in military ways. Inasmuch as widespread interest in military and commercial aeronautics exists, an elaborate program has been arranged for this meeting which affords an excellent opportunity for discussions of the technical problems necessarily involved, together with the military and economic questions. Second Vice-president Glenn L. Martin, representing aviation engineering, will preside at the technical session.

The papers for the technical session are particularly comprehensive in the treatment of the different phases of aircraft and navigation problems. It is expected that abstracts only will be given at the meeting, thus affording an opportunity for the discussion of the papers that probably will be printed and distributed prior to the meeting. Lieut.-Col. V. E. Clark will open the discussion of his paper entitled Possible Airplane Performance with Maintenance of Engine Power at All Altitudes. This paper contains interesting data by which possible speeds can be rapidly calculated for airplanes by the use of curves previously developed from the craft characteristics. Another paper equally interesting to engineers interested in civil aeronautics is Consideration of Landing Run and Get-Away by Standard Type Airplanes by Alexander Klemin. This paper outlines and indicates possible methods of braking an airplane in landing. An excellent paper by S. R. Parsons entitled Some Factors in the Design of Airplane Radiators contains the results of experiments and tests made at the Bureau of Standards on air flow through radiator tubes.

The Heat Treating of Brazed Fittings for Aircraft by Archibald Black, engineer in charge of aeronautical specifications, Bureau of Construction and Repair, Navy Department, Washington, is a timely paper on the brazing of fittings, specifying ways and means of better brazing and more particularly the means of procuring brazed heat-treated fittings.

Flying an Aviation Engine on the Ground by S. W. Sparrow, gives the data obtained from experiments and tests made at the Bureau of Standards with engines under altitude conditions in the laboratories at the Bureau. It is expected that the members will come prepared to discuss the papers presented and that statistics and other data of value from these discussions will supplement the papers.

In addition, a brief report from the Aeronautic Division of the Standards Committee of the Society will be submitted, with remarks by the chairman of the Division, M. H. Crane, on its past, present and future work.

The dinner at the Hotel Astor will be preceded by a reception where a chance for extending acquaintanceship is afforded. After-dinner speaking will include several phases of the economic requirements of aircraft engineering, military, naval and civil. Among the speakers will be Major-General C. T. Menoher, director of the Air Service; Capt. T. T. Craven, U. S. N., director of Air Operations, and George H. Houston and Frank H. Russell, who will speak on the commercial future of aeronautics. The presiding officer at the dinner will be President J. G. Vincent. It is expected the addresses on military, naval and civil aeronautics will lead to forecasts which will have a wholesome effect upon the industry and a lasting good to the country as a whole.

HERBERT CHASE RETURNS TO ENGINEERING PRACTICE

HERBERT CHASE, who for over 3 years has held the position of Assistant Secretary of the Society, has accepted a position with the Power Plants Corporation, a company organized to undertake engine development and other work of related character, a line of endeavor in which Mr. Chase has long been interested,

Mr. Chase is widely acquainted and favorably known in automotive circles. He has contributed frequently to the *Transactions* of the Society papers and discussions on engines and engine testing. It is understood that the engines he expects to develop will be of a type capable of utilizing the cheaper grades of fuel and that they will probably operate on the constant-pressure cycle.

During the war Mr. Chase did excellent work while in charge of the Washington office of the Society. He served also as a member of the Committee on Powerplants for Aircraft of the National Advisory Committee for Aeronautics. He has held membership on numerous committees of the Society, having for 3 years much to do with the arrangements for the professional papers and the meetings and doing very faithful and effective work in this connection. He was Treasurer of the Society in 1917.

STANDARDS COMMITTEE MEETING

(*Concluded from page* 6)

(12) *Engine Testing Forms*

As the present S. A. E. Standard for Engine Testing Forms is satisfactory for kerosene fuel engine testing, the Division has recommended that a note be added to the forms stating that they are intended for engines using either gasoline or kerosene fuel.

Iron and Steel Division

(13) *Tungsten Steel*

The Iron and Steel Division has recommended for adoption as S. A. E. Standard the following composition for low-tungsten steel, to be known as S. A. E. Specification No. 7260. Low-tungsten steel is more widely used for both inlet and exhaust valves than high-tungsten steel. The composition proposed conforms to Specification No. W60b of the Bureau of Aircraft Production.

Specification No. 7260

	Per cent
Carbon	0.50 to 0.700
Manganese, not to exceed	0.300
Phosphorus, not to exceed	0.035
Sulphur, not to exceed	0.035
Chromium	0.50 to 1.000
Tungsten	1.50 to 2.000

(14) *Nickel Steel*

As 5 per cent nickel steel is widely used for case-hardened parts, the Division has recommended for adoption as S. A. E. Standard the following composition for 5 per cent nickel steel to be known as S. A. E. Specification No. 2512.

Specification No. 2512

	Per cent
Carbon, not to exceed	0.170
Manganese	0.30 to 0.600
Sulphur, not to exceed	0.045
Phosphorus, not to exceed	0.040
Nickel	4.50 to 5.250

(15) *Malleable Iron Castings*

Malleable iron castings are being used to a large extent in the automotive industry, and the Division has therefore recommended that the A. S. T. M. Specification No. A47-19 for malleable castings be adopted as S. A. E. Standard.

In 1914 the original malleable castings specification adopted by the Society was withdrawn, as it was based on chemical analysis and was not practicable. The present proposal applies to the physical characteristics of malleable castings, which is considered the best method of controlling the quality of this product.

Malleable Iron Castings[1]

Material Covered These specifications cover malleable castings for general automotive purposes.

Process The castings shall be produced by either the air-furnace, open-hearth, or electric-furnace process.

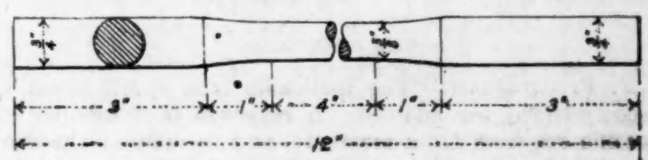
Tension Tests The tension test-specimens specified shall conform to the following minimum requirements as to tensile properties:

Tensile strength, lb. per sq. in.	45,000
Elongation in 2 in., per cent	7.5

Special Tests All castings, if of sufficient size, shall have cast thereon test-lugs of a size proportional to the thickness of the casting, but not exceeding 5/8 by 3/4 in. in cross-section. On castings which are 24 in. or over in length, a test-lug shall be cast near each end. These test-lugs shall be attached to the casting at such a point that they will not interfere with the assembling of the castings, and may be broken off by the inspector.

If the purchaser or his representative so desires, a casting may be tested to destruction. Such a casting shall show good, tough malleable iron.

Tension Test-Specimens[2] Tension test-specimens shall be of the form and dimensions shown. Specimens whose mean diameter at the smallest section is less than 19/32 in. will not be accepted for test.



Malleable Iron Tension Test-Specimen

A set of three tension test-specimens shall be cast from each melt, without chills, using heavy risers of sufficient height to secure sound bars. The specimens shall be suitably marked for identification with the melt. Each set of specimens so cast shall be placed in some one oven containing castings to be annealed.

Number of Tests After annealing, three tension test-specimens shall be selected by the inspector as representing the castings in the oven from which these specimens are taken.

If the first specimen conforms to the specified requirements, or if, in the event of failure of the first specimen, the second and third specimens conform to the requirements, the castings in that oven shall be accepted, except that any casting may be rejected if its test-lug shows that it has not been properly annealed. If either the second or third specimen fails to conform to the requirements, the entire contents of that oven shall be rejected.

Reannealing Any castings rejected for insufficient annealing may be reannealed once. The reannealed castings shall be inspected and if the remaining test-lugs, or castings broken as specimens, show the castings to be thoroughly annealed, they shall be accepted; if not, they shall be finally rejected.

Workmanship The castings shall conform substantially to the patterns or drawings furnished by the purchaser, and also to gages which may be specified in individual cases. The castings shall be made in a workmanlike manner. A variation of 1/8 in. per ft. will be permitted.

Finish The castings shall be free from injurious defects.

Marking The manufacturer's identification mark and

[1]This specification conforms with Specification No. A47-19 of the American Society for Testing Materials.

[2]Test-specimens are not machined.

the pattern numbers assigned by the purchaser shall be cast on all castings of sufficient size, in such positions that they will not interfere with the service of the castings.

Inspection The inspector representing the purchaser shall have free entry at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the castings ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the castings are being furnished in accordance with these specifications. All tests and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

The manufacturer shall be required to keep a record of each melt from which castings are produced, showing tensile strength and elongation of test-specimens cast from such melts. These records shall be available and shown to the inspector whenever required.

Rejection Castings which show injurious defects subsequent to their acceptance at the manufacturer's works may be rejected, and, if rejected, shall be replaced by the manufacturer free of cost to the purchaser.

THE DISCUSSION

F. P. GILLIGAN:—The malleable iron specification is recommended for adoption in response to a number of specific requests from members, and in view of the fact that the American Society for Testing Materials had developed the best type of malleable iron specifications that we knew of, the members of the Division felt that it would be desirable to adopt it without change.

CHAIRMAN B. B. BACHMAN:—There is a question of policy which should be considered in connection with this matter. The question has been raised as to whether it is the function of the S. A. E. to deal with matters having to do with the testing of materials. The proposal deals principally with the properties which shall be required and with the methods for determining these properties.

H. M. CRANE:—I feel that it would be wise to publish this proposal as general information and as a specification of the American Society for Testing Materials, as it involves the relations between purchaser and seller to a large extent. It brings in questions of inspection, etc., which I do not think properly belong in this Society's Standards as such. It is, however, a very valuable specification to have in the Handbook as to what has been worked out in a practical way toward determining whether malleable castings are satisfactory. It is generally acknowledged that it is very difficult to specify what the castings should be and a specification such as proposed will serve as a valuable guide to engineers. However, if the American Society for Testing Materials should alter its specifications we would have to do likewise.

MR. GILLIGAN:—The point in connection with malleable iron is that chemical composition alone is of no value. The maker must, of course, work within certain limits, but the castings may have the required composition and still lack the physical properties and other characteristics desired in malleable iron castings. The only way to control the quality of malleable iron is by making fracture tests, etc., leaving the composition of the castings to the maker. The Society had formerly a specification that was based primarily on composition, but it failed to attain its purpose. The proposed specification has been developed by specialists and represents the best American practice.

H. S. PIERCE:—It appears to me that the specification proposed is a broad generality. Anyone making many malleable castings soon learns that castings for various purposes require different degrees of hardness with reference to elongation, and that if malleable iron is not exactly one substance, it can be made to suit conditions.

H. J. STAGG:—I think that the main point to be brought out here is that sometimes a chemical specification means absolutely nothing. The physical specification is all-important. If the Society adopts a malleable iron specification, I am strongly in favor of adopting a method of determining the physical properties of the iron.

MR. CRANE:—The conditions applying to malleable iron are the same as apply to cast iron and to the method of making piston-rings. The specification of chemical analysis alone is an absolutely impossible method of controlling the quality of castings.

G. L. NORRIS:—I believe that it should be the policy of the Society to have its specifications conform with those of the American Society for Testing Materials in respect to the methods of testing and determining physical properties.

MR. CRANE:—With respect to the test-specimen in this specification, is it to be understood that the bars are not machined?

JAMES S. MACGREGOR:—It is important in a specification of this kind to state the condition of the test-specimen. Specimens of malleable cast iron tested as cast and malleabilized yield very different results from those obtained if the specimens are machined before testing.

MR. GILLIGAN:—It is not customary to machine these test-specimens. A footnote to this effect can be included in the specification.

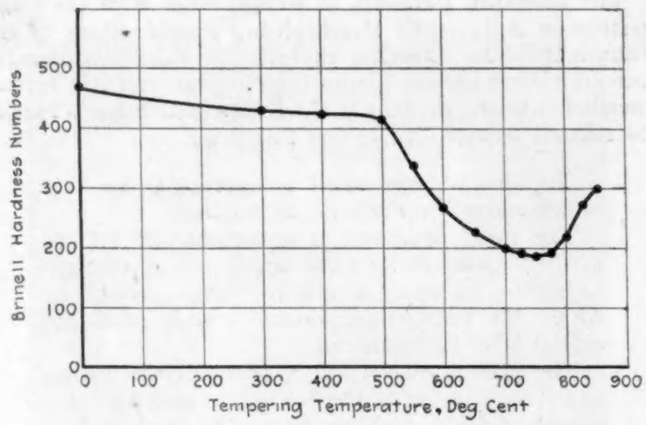
(16) *High-Chromium Steel*

In line with the Division policy of publishing in THE JOURNAL and the S. A. E. Handbook information about all types and uses of new steels coming before the industry, a Sub-Division was appointed to review all available data in reference to high-chromium steel. The following report has been accepted by the Division and published in THE JOURNAL.

High-chromium, or what is called stainless steel containing from 11 to 14 per cent chromium, was originally developed for cutlery purposes, but has in the past few years been used to a considerable extent for exhaust valves in airplane engines because of its resistance to oxidation or scaling at high temperatures.

	Per cent
Carbon	0.20 to 0.400
Manganese, not to exceed	0.500
Phosphorus, not to exceed	0.035
Sulphur, not to exceed	0.035
Chromium,	11.50 to 14.000
Silicon, not to exceed	0.300

Forging The steel should be heated slowly and forged at a temperature above 1750 deg. fahr., preferably between 1800 and 2200 deg. fahr. If forged at temperatures between 1650 and 1750 deg. fahr. there is considerable danger of rupturing the steel because of its hardness at red heat. Owing to the air-hardening property of the steel, the drop-forgings should be trimmed



RELATION BETWEEN THE TEMPERING TEMPERATURES AND THE BRINELL HARDNESS NUMBERS OF HIGH-CHROMIUM STEEL

while hot. Thin forgings should be reheated to redness before trimming, as otherwise they are liable to crack.

Heat-Treatment Forgings will be hard if they are allowed to cool in air. This hardness varies over a range of from 250 to 500 Brinell, depending on the original forging temperature.

Annealing This can be done by heating to temperatures ranging from 1290 to 1380 deg. fahr. and cooling in air or quenching in water or oil. After this treatment the forgings will have a hardness of about 200 Brinell and a tensile strength of 100,000 to 112,000 lb. per sq. in. If softer forgings are desired they can be heated to a temperature of from 1560 to 1650 deg. fahr. and cooled very slowly. Although softer the forgings will not machine as smoothly as when annealed at the lower temperature.

Hardening The forgings can be hardened by cooling in still air or quenching in oil or water from a temperature between 1650 and 1750 deg. fahr.

Valves These have generally been made to the following specification of physical properties:

Yield point, lb. per sq. in.	70,000
Tensile strength, lb. per sq. in.	90,000
Elongation in 2 in., per cent	18
Reduction of area, per cent	50

The usual heat-treatment is to quench in oil from 1650 deg. fahr. and temper or draw at 1100 to 1200 deg. fahr. One valve manufacturer stated that valves of this steel are hardened by heating the previously annealed valves to 1650 deg. fahr. and cooling in still air. This treatment gives a scleroscope hardness of about 50.

Cold-Working This steel can be drawn into wire, rolled into sheets and strips and drawn into seamless tubes.

Corrosion This steel like any other steel when distorted by cold working is more sensitive to corrosion and will rust. Rough cut surfaces will rust. Surfaces finished with a fine cut are less liable to rust. Ground and polished surfaces are practically immune to rust.

Scaling Comparative resistance to scaling or oxidation at high temperatures is shown in the accompanying chart.

Physical Properties The physical properties do not vary greatly when the carbon is within the range of

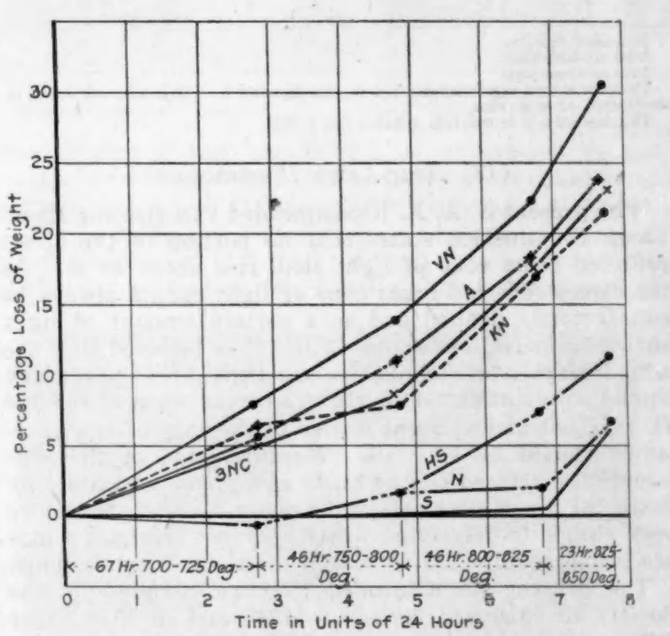
COMPARISON OF PHYSICAL PROPERTIES FOR HIGH-CHROMIUM STEELS OF DIFFERENT CARBON CONTENT

	C 0.20, Mn 0.45, Cr 12.56	C 0.27, Mn 0.50, Cr 12.24	C 0.50, Cr 14.84
Quenched in oil from deg. fahr.	1,600	1,600	1,650
Tempered at deg. fahr.	1,160	1,080	1,100
Yield point, lb. per sq. in.	78,300	75,000	91,616
Tensile strength, lb. per sq. in.	104,600	104,250	123,648
Elongation in 2 in., per cent	25.000	23.500	14.500
Reduction of area, per cent	52.500	51.400	33.500

COMPARISON OF PHYSICAL PROPERTIES BETWEEN AIR, OIL AND WATER-HARDENED STEEL HAVING CHEMICAL ANALYSIS IN PERCENTAGE OF

Carbon	0.240
Manganese	0.300
Phosphorus	0.035
Sulphur	0.035
Chromium	12.850
Silicon	0.200

Hardening Medium	Hardened from. deg. fahr	Tempered at, deg. fahr.	Elastic Limit, lb. per sq. in	Tensile Strength, lb. per sq. in.	Elongation in 2 In., per cent	Reduction of Area, per cent
Air	1,650	930	158,815	192,415	13.0	40.5
		1,100	99,680	120,065	21.0	59.2
		1,300	70,785	101,250	26.0	64.6
		1,380	66,080	98,335	28.0	63.6
		1,470	70,785	96,990	27.0	64.7
Oil	1,650	930	163,070	202,720	8.0	18.2
		1,100	88,255	116,480	20.0	56.9
		1,300	77,950	105,505	25.5	63.8
		1,380	88,255	98,785	27.0	66.8
Water	1,650	930	158,815	202,050	12.0	34.2
		1,100	90,270	120,735	22.0	59.6
		1,300	66,080	102,590	25.8	64.8
		1,380	67,200	97,890	27.0	65.2



S, high-chromium steel; HS, high-speed steel; KN, 3NS, nickel chromium steel; N, 25 per cent nickel steel; N, 5 per cent nickel steel; A, 0.30 per cent carbon steel.

Specimens were heated at the temperatures stated and weighed every 24 hr. after removing the scale.

Temperatures in degrees centigrade.

RESULTS OF SCALING TESTS OF VARIOUS STEELS

composition given, or when the steel is hardened and tempered in air, oil, or water.

Applications In addition to use in valves this steel should prove very satisfactory for shafting for water-

pumps and other automobile parts subject to objectionable corrosion.

THE DISCUSSION

CHAIRMAN B. B. BACHMAN:—This proposal is presented for consideration as general information only. The reason for this is that it is claimed that certain patent rights apply to this particular steel, and in accordance with our procedure we are not adopting as standard or recommended practice anything claimed to be of a proprietary nature.

LIGHTING DIVISION

(17) *Side-Lamp Glasses*

The Lighting Division has recommended for adoption as S. A. E. Standard the following series of side-lamp glass dimensions. The present standard tail-lamp glass is 3 in. in diameter with tolerances of plus 1/32 in., minus 1/64 in. The recommendation changes these tolerances to conform with those recommended for head-lamp, side-lamp and spot-lamp glasses. The recommendation, together with the present standard for head-lamp glasses is intended for automobile electric and oil-lamp glasses including types such as semaphore, beehive and prism.

DIMENSIONS FOR SIDE-LAMP GLASSES

OUTSIDE DIAMETER OF LENS		MINIMUM LIGHT OPENING IN DOOR	
Nominal Diameter	Tolerances	Nominal Diameter	Tolerances
2	+0, −1/32	1¾	+0, −1/32
2½	+0, −1/32	2⅛	+0, −1/32
*3	+0, −1/32	2½	+0, −1/32
5	±1/32	4	±1/32
†6	±1/32	5	±1/32
7	±1/32	6	±1/32

Dimensions in inches.
*Also tail-lamp glass.
†Also spot-lamp glass.
Thickness of bezel edge for all sizes is ⅛ in., plus 1/32, minus 0. This is commonly known as double-thick American glass.
This standard is to become fully operative July 1, 1921.

(18) *Head-Lamp Illumination*

The present S. A. E. Recommended Practice for Head-Lamp Illumination states that no portion of the direct reflected beam cone of light shall rise above 42 in. As the direct reflected beam cone of light cannot always be satisfactorily defined and as a certain amount of light should be permitted above 42 in., it is believed that the actual glare and illumination candlepower values which should obtain at certain parts of a screen when placed 100 ft. in front and at right angles to the axis of the head-lamps should be specified. Measurements of the illumination produced can be made easily and accurately by means of a portable foot-candle meter and it is therefore very simple to determine whether or not the head-lamps are properly adjusted to comply with any requirements.

The present Recommended Practice, adopted by the Society in 1916 and revised in 1917 and in 1918, is as follows:

> The head-lamps shall be arranged so that no portion of the direct reflected beam cone of light when measured 75 ft. ahead of the head-lamps shall rise above 42 in. from the level surface of the road on which the vehicle stands under any conditions of loading; nor shall any portion of the direct reflected beam cone of light rise beyond the 75-ft. distance more than 12 in. above the center of the head-lamps.

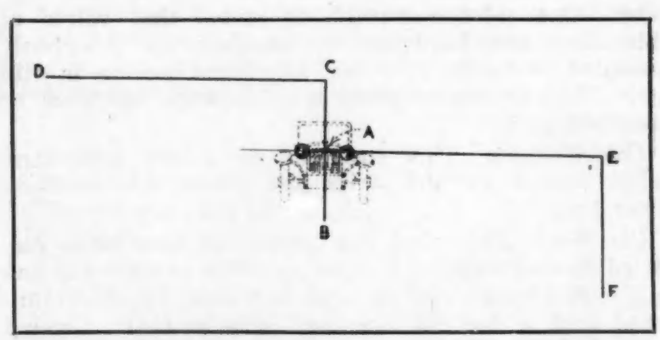
The Lighting Division, in co-operation with the Committee on Automobile Headlighting Specifications of the Illuminating Engineering Society, has done considerable research work on head-lamp illumination and has recommended that the present S. A. E. Recommended Practice be revised to conform to the following:

> The head-lamps shall be arranged so that under normal conditions of loading
>
> The light produced at a distance of 100 ft. directly ahead of the head-lamps and at a height of 60 in. or more above the level surface on which the automotive vehicle stands shall not exceed 2400 apparent cp.
>
> The light produced 100 ft. ahead of the vehicle and 7 ft. or more to the left of its axis and at a height of 60 in. or more above the level surface on which the vehicle stands shall not exceed 800 apparent cp.

This recommendation is based on the results of a series of tests made in 1918 and 1919. The first tests were made to indicate how much light is necessary to reveal a man in dark clothes 250 ft. in front of a car and how much light the driver can tolerate without feeling that the glare is objectionable and unsafe to drive against. Head-lamps were placed on a level concrete road in relatively the same positions that they would be on two cars preparing to pass, the distance between the two pairs of head-lamps being 100 ft.

The electrical and photometric equipment was such that the intensity of the light for either pair of head-lamps could be varied by the observer and his opinions of glare and illumination recorded. The procedure followed was to have the observer increase the illumination of his own head-lamps until he felt that the illumination was adequate to reveal a man at 150 and then at 250 ft. in front of the head-lamps. Then with these head-lamps set at this recorded intensity the opposing head-lamps were turned on and the illumination increased until in the observer's opinion the glare was at a maximum for safe and convenient driving.

Subsequent tests were made with automobiles equipped with head-lamps adjusted to give approximately the values deduced from the observations made in the first tests by the fifty observers. It was decided as a result of these tests that it was necessary to have approximately 4800 apparent cp. to obtain satisfactory road illumination at 200 ft. and that dangerous glare would be experienced



Location of points is as follows: *A* is level with center of head-lamps and 100 ft. ahead of vehicle, *B* is 1 deg. or 21 in. below *A*, *C* is 1 deg. or 21 in. above *A*, *D* is 4 deg. or 7 ft. to the left of *C* as observed from the vehicle, *E* is 4 deg. or 7 ft. to the right of *A*, and *F* is 2 deg. or 42 in. below *E*. The passenger car is shown in the illustration only to indicate the location of point *A* with respect to the head-lamps.

POINTS USED FOR SPECIFYING CANDLEPOWER VALUES

TYPE OF PORTABLE FOOT-CANDLE METER USED

if the illumination at 1 deg. above the horizontal exceeded 2400 apparent cp. and the illumination at 1 deg. above and 4 deg. to the left of the axis of the car exceeded 800 apparent cp.

Tests were recently conducted with head-lamps adjusted to give the following illumination and glare candlepower values at 100 ft. ahead of the head-lamps and for the points shown in the drawing on page 60. The candlepower values were determined by the foot-candle meter shown.

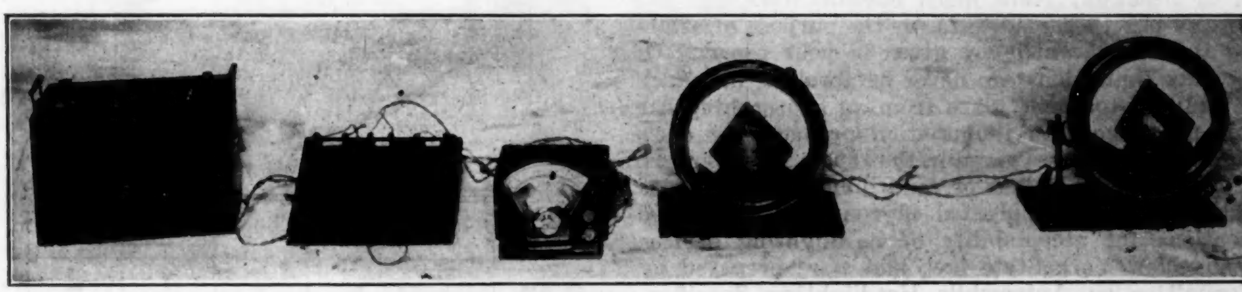
At some point between *A* and *B*, 4800 apparent cp.
At point *C*, 2400 apparent cp.
At point *D*, 800 apparent cp.
At some point between *E* and *F*, 1200 apparent cp.

The equipment consisted of three cars equipped with special head-lamps, lenses, storage batteries, voltmeters and rheostats. The equipment for each car was as shown on this page. The cars were operated on a dark night at equal intervals on a level macadam road recently surfaced, which was selected on account of the absence of street and house lights. After 20-min. runs observers were requested to give their opinions as to the glare produced by the head-lamps of oncoming cars and the adequacy of the illumination produced on the road by the head-lamps of the car in which they were riding.

The result of the tests showed that ten observers were of the opinion that the glare was excessive, ten that it was not excessive, and six that it was the most that should be permitted. The analysis of the observations, however, indicated that the glare value was satisfactory as a maximum.

The observations also indicated that the illumination of 4800 apparent cp. as a minimum at some point between *A* and *B* was satisfactory as a legal requirement, but that a higher value should be specified for recommended practice. It is thought that an illumination of 10,000 cp. at some point between *A* and *B* would be more satisfactory and that the light produced at 4 deg. on either side of the axis of the vehicle and between the level of the head-lamps and 2 deg. below this level should be increased to at least 2400 cp.

The Division in cooperation with the Illuminating Engineering Society's Committee plans to run additional tests with head-lamps calibrated to give these proposed candlepower values.

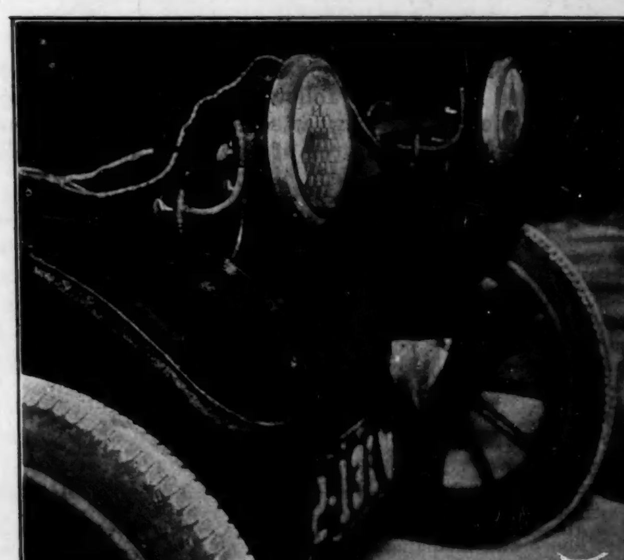


EQUIPMENT USED ON EACH TEST PASSENGER CAR.

THE DISCUSSION

H. M. CRANE:—The Lighting Division in cooperation with the Illuminating Engineering Society has done much toward working out an arrangement that can be readily controlled and which will give the best possible driving light combined with the least possible amount of dangerous glare. There are undoubtedly patented head-lamp glasses which would do the work perfectly well if applied to the car properly, but we frequently see cars with various types of lenses which have worked out of their proper position, and instead of lighting the road they throw a strong light in the eyes of the oncoming driver. One object of the committee is to develop, if possible, some type of control that can be readily understood by the average owner and which can be expected to continue to perform the desired service without continuous attention.

W. A. MCKAY:—The former Recommended Practice of the Society was that the head-lamp should be arranged so that no portion of the direct reflected beam cone of



METHOD OF MOUNTING SPECIAL HEAD-LAMPS.

light should rise above 42 in. from the surface of the road. As the direct reflected beam cone of light cannot always be satisfactorily defined and as a certain amount of light should be permitted above 42 in., it is believed that the actual glare and illumination candlepower values which should obtain at certain parts of a screen when placed 100 ft. in front of, and at right angles to, the axis of the head-lamps should be specified. Measurements of the illumination produced can be easily and accurately made by a portable foot-candle meter and it is therefore very simple to determine whether or not the head-lamps are properly adjusted to comply with any requirements. Formerly head-lamp specifications simply stated that certain conditions should not exist, usually that there should not be a certain glare. There was always difficulty in determining whether the head-lamps did produce dangerous glare. The proposed specifications have definite candlepower values so that the illumination can be measured with a portable foot-candle meter, making it possible for those who have to do with the enforcement of laws to find out definitely whether a head-lamp is adjusted in accordance with such a specification as the one proposed.

DR. C. H. SHARP:—The Committee on Automobile Headlighting Specifications of the Illuminating Engineering Society has given the matter of the proper limitation of headlight glare much consideration and has made many practical road tests for the purpose of establishing the values which are given in your report. Of course glare limitations can never be absolute, inasmuch as the limit of tolerable glare from an approaching car depends upon the road illumination produced by one's own lights. The figures given in the report, however, are based on reasonable values of road illumination. These figures were first promulgated in connection with the specifications for acceptability of headlighting devices under the New York State law, and were prepared by the Committee on Automobile Headlighting Specifications at the request of Secretary of State Hugo for that purpose. Since then they have entered into the regulations of the States of Connecticut, Pennsylvania and California. They therefore have already a very considerable degree of sanction on the part of the law-making and law-administrating authorities.

MISCELLANEOUS DIVISION

(19) *Oilless Bushings*

The Division has recommended for adoption as S. A. E. Recommended Practice the following sizes of oilless bushings. The subject was suggested as suitable for standardization by a manufacturer of bearings, as practically every automobile builder specifies different size bushings, necessitating making these parts to order. Data were obtained from the industry showing sizes of bushings in use and the attendant recommendation represents a great reduction in the number of sizes.

[The title of this subject was originally Brake-Shaft Bushings.]

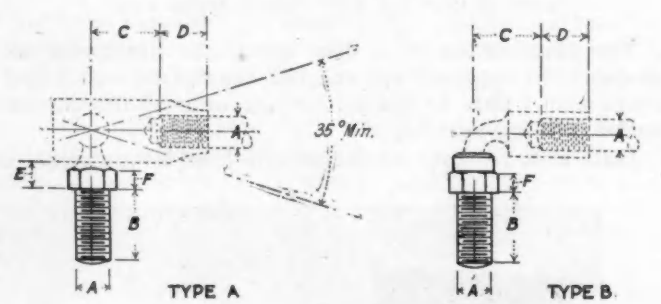
DIMENSIONS FOR OILLESS BUSHINGS

Inside Diameter[a]	Length		Inside Diameter[a]	Length	
1/4	3/16	1/4	7/8	21/32	7/8
5/16	1/4	5/16	15/16	11/16	15/16
3/8	9/32	3/8	1	3/4	1
7/16	11/32	7/16	1 1/8	7/8	1 1/8
1/2	3/8	1/2	1 1/4	15/16	1 1/4
9/16	7/16	9/16	1 3/8	1	1 3/8
5/8	15/32	5/8	1 1/2	1 1/8	1 1/2
11/16	1/2	11/16	1 5/8	1 1/4	1 5/8
3/4	9/16	3/4	1 3/4	1 5/16	1 3/4
13/16	5/8	13/16	2	1 1/2	2

Dimensions in inches.
Wall thickness: below 1¼ in. inside diameter, 1/32, 1/16, and ⅛ in.; above and including 1¼ in. inside diameter, 1/32, 1/16, ⅛ and 3/16 in.
[a]Inside diameter tolerances: below 1 in., + 0.001 or + 0.003 in.; above and including 1 in., + 0.001 or + 0.004 in.

(20) *Ball-and-Socket Joints*

The Division has recommended for adoption as S. A. E. Standard the attendant dimensions for ball-and-socket joints. The tentative recommendation has been submitted to manufacturers and users and their criticism obtained. The proposed standard applies to the two most widely used types of ball-and-socket joints and the series of sizes proposed meets all general requirements for spark and throttle control.



DIMENSIONS FOR BALL-AND-SOCKET JOINTS

A	Hex Min	B	C	D	[4]E	[5]F Min
No. 5-40	1/4	9/32	1/4	1/4	7/32	3/32
No. 8-32	1/4	11/32	11/32	5/16	7/32	7/64
No. 10-32	5/16	13/32	13/32	3/8	1/4	1/8
No. 12-32	3/8	15/32	7/16	3/8	1/4	9/64
1/4-28	3/8	9/16	1/2	7/16	9/32	5/32
5/16-24	7/16	21/32	5/8	9/16	9/32	3/16
3/8-24	9/16	7/8	3/4	11/16	5/16	7/32
7/16-20	5/8	1	7/8	13/16	3/8	1/4
1/2-20	3/4	1 1/8	1	15/16	7/16	9/32

[4]Minimum wrench clearance.
[5]F = ½ A + 1/32.
Dimensions in inches.
All threads are S.A.E.

[Referring to the table the following changes were made: In column A No. 12-32 size was 7/32-32. In column B 9/32 was ¼, 13/32 was ⅜ and 15/32 was 7/16. In the last column of the table 9/64 was 5/32 and 9/32 was 5/16.]

THE DISCUSSION

E. H. EHRMAN:—The standardization of ball-and-socket joints has been under consideration by the Miscellaneous Division for approximately 2 years and the members believe that the proposed specification will cover practically all requirements in a satisfactory manner. There is one point that I would like to explain. The National Screw Thread Commission plans to discard all fractional sizes below ¼ in., so the Division recommends that 7/32 in. be expressed as No. 12, since it is almost identical with it. The pitch of 32 threads per inch is pretty well established in the product of one manufacturer of ball-and-socket joints and it is claimed that the sale of this size of joint is very extensive, so that it might work a hardship to change. The S. A. E. adopted

in aircraft work, 28 threads per inch for the No. 12 size, and at the time of its adoption the criticism was that manufacturers could not buy from the tap makers No. 12 taps having 32 threads. It is apparent, however, from circularizing the manufacturers and users of ball-and-socket joints that the 32-pitch thread is satisfactory and that there apparently is no difficulty in obtaining No. 12-32 taps and dies.

ALEX TAUB:—I would like to find out just where a ½-in. ball-and-socket joint could be used aside from drag-links. Why standardize it?

MR. EHRMAN:—The series was purposely extended to include the largest and smallest sizes proposed. It is difficult to determine the limits of the range of sizes, but to my personal knowledge the proposed largest and smallest sizes have been used; for what purposes, however, I do not know. Since there is a demand for them the Division believes a standard should be established as a guide.

CHAIRMAN B. B. BACHMAN:—I think that the answer to Mr. Taub's question is that the Society is dealing with not only passenger cars but trucks, tractors, marine apparatus, motorcycles, stationary internal-combustion engines and airplanes. This expands the field of our work to such an extent that the specifications of the Standards Committee to be complete must take into consideration requirements of all these different fields of automotive work. There are many of us who being interested in one particular line naturally may not see the reason for the extension of these standards outside of our own experience.

MR. TAUB:—Is it not a fact nevertheless that these larger sizes cannot be purchased from stock, but must be ordered specially?

MR. EHRMAN:—To illustrate the reason for the proposal, I will refer to the S. A. E. Standard screws. The sizes that are used most are carried in stock and the extreme lengths and the larger sizes have to be made to order. No one would be expected to carry 1½-in. S. A. E. screws in stock because there is so little call for them; still the standard covers this size.

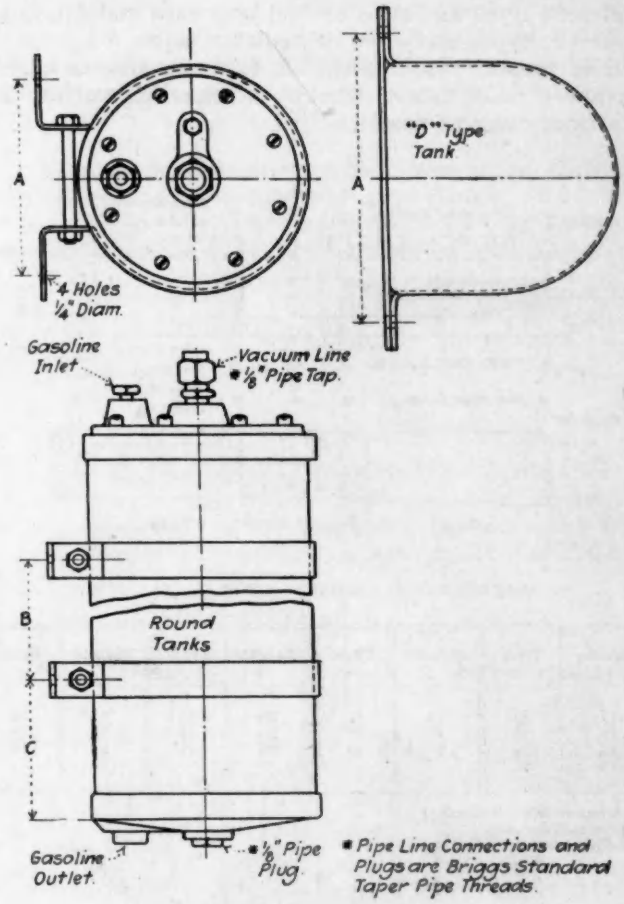
F. C. GOLDSMITH:—The idea in this matter is to work out a suitable standard and have it adopted by the Society without waiting until individual practices become established.

(21) *Fuel Vacuum-Tank Mounting and Connections*

This subject was proposed for standardization at the 1919 Summer Meeting of the Society. It was stated at that time that the vacuum method of regulating the flow of gasoline to the carbureter has been adopted for use on approximately 80 per cent of the different makes of automobiles, but that there was no standard practice for the mounting dimensions or connections for the fuel and vacuum-pipe lines.

It will be noted that the pipe-line connections for the vacuum pipes and the gasoline inlets and outlets are specified as Briggs standard taper pipe threads and conform to S. A. E. Recommended Practice for carbureter fittings, pages 35a and 35b, S. A. E. Handbook, Vol. I.

The Miscellaneous Division has therefore recommended for adoption as S. A. E. Standard the fuel vacuum-tank mountings and pipe conections shown in the accompanying drawing and table.



FUEL VACUUM-TANK MOUNTINGS

FUEL VACUUM-TANK MOUNTINGS AND CONNECTIONS

Tank Diameter, In.	* Gasoline Inlet	* Gasoline Outlet	A	B	C
3¾ and 4½	⅛	⅛	4	4	2½
6	⅛	¼	4	4	2½
"D" type	¼	¼	7¼	4	2½

All dimensions are in inches.
*Pipe-line connections and plugs are Briggs standard taper pipe threads.

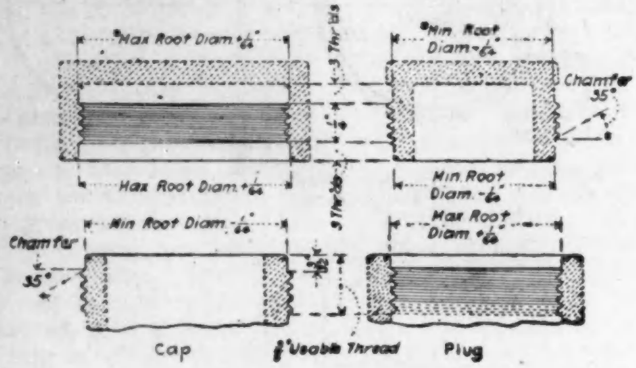
(22) *Tank and Radiator Caps*

The Division has recommended for adoption as S. A. E. Recommended Practice the following cap and plug types of tank and radiator caps. No thread tolerances were proposed by the Division, as it is desired to base them on the final report of the National Screw Thread Commission. A recess of three threads is specified for the cap in order to allow sufficient clearance for tapping.

This proposal is based on the recommendation made at the June, 1919, meeting of the Standards Committee, when it was referred back to the Division for further consideration. It was the consensus of opinion that the S. A. E. Standard fine threads should be specified rather than 16 U. S. Standard pitch for all sizes and that the ⅛-in. sizes should be eliminated. The original recommendation was based on data obtained from fifty-six companies representing both makers and users, and showed that there were in use by those companies sixty

different types and sizes of fuel tank caps and fifty-eight different types and sizes of radiator caps.

The present recommendation is made after a careful review of automobile, tractor, aeronautic, marine, and stationary engine practice.



DIMENSIONS FOR RADIATOR AND FUEL TANK CAPS

Diameter	Pitch	Diameter	Pitch	Diameter	Pitch	Diameter	Pitch
1	20	1½	18	2¼	16	3	16
1⅛	18	1¾	16	2½	16	3½	16
1¼	18	2	16	2¾	16	4	16
1⅜	18						

* Recess or neck is optional.
Corner radii are 1/16 in.
All threads are S. A. E. Fine.

Motorcycle Division

(23) *Spark-Plug Shells*

TAPPED HOLE DIMENSIONS

Diameter	Maximum		Minimum	
	Mm.	In.	Mm.	In.
Outside (full)	18.265	0.7191	18.115	0.7171
Pitch (effective)	17.127	0.6743	17.076	0.6723
Root (core)	16.152	0.6359	16.101	0.6339

SPARK-PLUG THREAD DIMENSIONS

Diameter	Maximum		Minimum	
	Mm.	In.	Mm.	In.
Outside (full)	18.001	0.7087	17.925	0.7057
Pitch (effective)	17.026	0.6703	16.949	0.6673
Root (core)	15.890	0.6256	15.839	0.6236

The Motorcycle Division has recommended that the thread limits for the present S. A. E. Recommended Practice for Spark-Plug Shells be extended to four decimal places and the millimeter equivalents included. The revision is proposed in order to provide a definite specification in accordance with good motorcycle engineering practice. This specification together with all other S. A. E. screw-thread standards will probably be more or less affected by the final report of the National Screw Thread Commission.

Non-Ferrous Metals Division

There have been considerable advances in the art of manufacturing aluminum castings since the adoption of the present aluminum alloy standards in 1911, and the Non-Ferrous Metals Division has therefore undertaken to bring these standards into conformity with the best aluminum casting practice of today.

Data have been obtained from manufacturers and users of non-ferrous metal alloys and considered by a subdivision. The Division has recommended that the present S. A. E. Standard for Aluminum Alloys, page 13, S. A. E. Handbook, Vol. I, be revised to conform to the following proposal which is based on the subdivision report.

(24) *Aluminum Alloy—Specification No.* 30

Aluminum, not less than, per cent	90.00
Copper, per cent	8.5 to 7.0

All other elements shall not exceed 1.7 per cent, of which not over 0.2 per cent shall be zinc. No other elements except silicon, iron, manganese and tin shall be allowed.

The tensile strength of test-specimens about ½ in. in diameter of this alloy cast in sand and tested without machining off the skin should be about 18,000 to 20,000 lb. per sq. in. and the elongation 1 to 2 per cent in 2 in.

This is a light alloy having a specific gravity of about 2.83 and is used more extensively in the automotive industry than all other light casting alloys combined. A shrinkage of 0.156 (5/32) in. per ft. should be allowed in pattern designs. This alloy is used for crankcases, oil-pans, steering-wheel spiders, differential carriers, transmission cases, camshaft housings, hub-caps and similar parts.

(25) *Aluminum Alloy—Specification No.* 31

Aluminum, not less than, per cent	81.00
Copper, per cent	3.25 to 2.25
Zinc, per cent	14.5 to 12.5

All other elements shall not exceed 1.7 per cent. No other elements except silicon, iron, manganese and tin shall be allowed.

The tensile strength of test-specimens about ½ in. in diameter of this alloy cast in sand and tested without machining off the skin should be about 25,000 to 30,000 lb. per sq. in. with an elongation of more than 1 per cent in 2 in.

The specific gravity is about 3.0 and a shrinkage of 0.156 (5/32) in. per ft. should be allowed in pattern designs.

This alloy is used extensively in England for such parts as crankcases, oil-pans, steering-wheel spiders and transmission cases.

(26) *Aluminum Alloy—Specification No.* 32

Aluminum, not less than, per cent	85.5
Copper, per cent	13.5 to 11.00

All other elements shall not exceed 1.7 per cent. The zinc shall not exceed 0.2 per cent. No other elements except silicon, iron, manganese and tin shall be allowed.

The tensile strength of test-specimens about ½ in. in diameter of this alloy cast in sand and tested without machining off the skin should be about 19,000 to 23,000

lb. per sq. in. and the elongation will be practically nothing.

The specific gravity of this alloy is about 2.95 and a shrinkage of 0.156 (5/32) in. per ft. should be allowed in pattern designs. This alloy is used for manifolds, pumps, carbureters, cylinders and other parts which should be free from leaks and where the brittleness of the alloy is not objectionable.

[The final sentence of the original recommendation reading "This alloy is also used extensively for pressure die castings" was deleted.]

THE DISCUSSION

R. W. WOODWARD:—Specification No. 32 is changed entirely from an aluminum-zinc alloy to an aluminum-copper alloy and the properties of the new alloy are outlined.

A MEMBER:—With respect to the use of these alloys for pressure die castings, I do not believe any manufacturer would undertake a contract with any of these specifications. Every automobile on the market today has anywhere from ten to fifty die castings and the die-casting process must therefore be considered. Neither these chemical nor physical specifications can be applied to any die casting that is being produced today. One company at least produces about 2,000,000 lb. of aluminum die castings per month. While I admit that the specification from a purely theoretical standpoint may be all right, the practical application of the present-day methods has not been considered sufficiently and I think the reference to the use of the specifications for pressure die castings should be omitted.

F. W. ANDREW:—I believe that the statement that these formulas are not good for die castings is correct and that they should not be passed for that purpose.

CHAIRMAN B. B. BACHMAN:—Is it to be understood, therefore, that these compositions are suitable for sand castings?

MR. WOODWARD: — I think these specifications were drawn up principally for use in making sand castings. The preparation of formulas for die-casting alloys will be taken up separately by the Division.

MARINE DIVISION

(27) *Reverse Couplings—Steel*

The present S. A. E. Standard for Steel Reverse Couplings has been criticised because the nuts specified do not conform to the S. A. E. Standard, causing unnecessary expense and trouble in obtaining special nuts for use in standard couplings. The Division has, therefore, proposed that the present standard be revised so as to allow the use of S. A. E. Standard nuts. The proposed nut thicknesses for thread diameters of over 1½-in. were determined according to the empirical formula for the smaller standard sizes: thickness = ⅞ of bolt diameter.

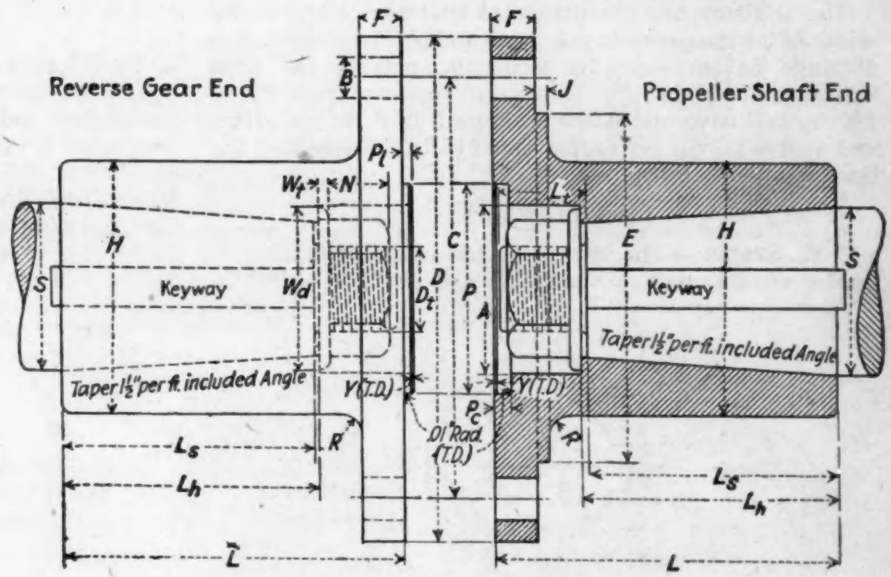
(28) *Water-Pipe Flanges*

Present marine practice is to use S. A. E. Standard carbureter flanges for water-pipe flanges. The Division has therefore recommended the following proposal for adoption as S. A. E. Recommended Practice:

> The present S. A. E. Standard for two and four-bolt type carbureter flanges given on data sheets 35 and 35xa, S. A. E. Handbook, Vol. I, is S. A. E. Recommended Practice for Water-Pipe Flanges.

(29) *Motorboat Control Levers*

The Marine Division feels that standardization of the direction of the movement of motorboat control levers will simplify the handling of motorboats and decrease the number of accidents caused by unfamiliarity with



MOTORBOAT STEEL REVERSE COUPLINGS.

DIMENSIONS FOR MOTORBOAT STEEL REVERSE COUPLINGS

Size No. (See Note)	Shaft and Nut[7]						Keyway		Washer		Flange					Pilot				Hub				Bolts, S.A.E.		
	S	Ls	Dt	N	[8]Thds. S.A.E.	Lt	Width	Height	Wd	Wt	D	E	F	J	A	[9]P	Pt	Pc	Y	H	L	Lh	R	No.	[10]B	C
RPS- 8	1	1 13/16	9/16	31/64	18	11/16	1/4	1/8	1 3/16	1/8	4	2 13/32	3/8	1/16	1 3/8	1 5/8	5/16	1/8	1/64	1 7/8	2 9/16	1 7/8	1/16	4	3/8	3
RPS- 9	1 1/8	1 13/16	9/16	31/64	18	11/16	1/4	1/8	1 3/16	1/8	4	2 13/32	3/8	1/16	1 3/8	1 5/8	5/16	1 1/8	1/64	1 7/8	2 9/16	1 7/8	1/16	4	3/8	3
RPS-10	1 1/4	2 3/16	3/4	21/32	16	7/8	5/16	5/32	1 7/16	1/8	5	3 13/32	7/16	3/32	1 3/4	2	3/8	3/16	1/32	2 1/4	3 1/8	2 1/4	1/32	6	7/16	4
RPS-11	1 3/8	2 3/16	3/4	21/32	16	7/8	3/8	3/16	1 7/16	1/8	5	3 13/32	7/16	3/32	1 3/4	2	3/8	3/16	1/32	2 1/4	3 1/8	2 1/4	1/32	6	7/16	4
RPS-12	1 1/2	2 9/16	7/8	49/64	16	7/8	3/8	3/16	1 7/16	1/8	5	3 13/32	7/16	3/32	1 3/4	2	3/8	3/16	1/32	2 1/4	3 1/8	2 1/4	1/32	6	7/16	4
RPS-13	1 5/8	2 9/16	7/8	25/32	16	7/8	3/8	3/16	1 7/16	1/8	5	3 13/32	7/16	3/32	1 3/4	2	3/8	3/16	1/32	2 1/4	3 1/8	2 1/4	1/32	6	7/16	4
RPS-14	1 3/4	2 15/16	1	7/8	14	1 1/8	7/16	7/32	1 13/16	1/8	6	4 1/16	1/2	1/8	2 3/16	2 1/2	3/8	3/16	1/32	3	4 1/8	3	1/8	6	9/16	5
RPS-15	1 7/8	2 15/16	1	7/8	14	1 1/8	1/2	1/4	1 13/16	1/8	6	4 1/16	1/2	1/8	2 3/16	2 1/2	3/8	3/16	1/32	3	4 1/8	3	1/8	6	9/16	5
RPS-16	2	2 15/16	1	7/8	14	1 1/8	1/2	1/4	1 13/16	3/16	6	4 1/16	5/8	1/8	2 3/16	2 1/2	3/8	3/16	1/32	3	4 1/8	3	1/8	6	9/16	5
RPS-18	2 1/4	3 11/16	1 1/8	63/64	12	1 5/16	5/8	5/16	2 7/16	3/16	7	4 1/2	5/8	1/8	2 1/2	3	7/16	1/4	1/32	3 3/4	5 3/16	3 3/4	1/16	6	5/8	5 1/2
RPS-20	2 1/2	3 11/16	1 1/8	63/64	12	1 5/16	5/8	5/16	2 7/16	3/16	7	4 1/2	5/8	1/8	2 1/2	3	7/16	1/4	1/32	3 3/4	5 3/16	3 3/4	1/16	6	5/8	5 1/2
RPS-22	2 3/4	4 1/8	1 1/4	1 3/32	12	1 5/8	5/8	3/8	2 13/16	1/4	8	5 3/8	3/4	1/8	3	3 3/4	1/2	5/16	1/64	4 3/8	5 7/8	4 3/8	1/8	6	3/4	6 1/2
RPS-24	3	4 3/8	1 3/8	1 13/64	12	1 5/8	3/4	3/8	2 13/16	1/4	8	5 3/8	3/4	1/8	3	3 3/4	1/2	5/16	1/64	4 3/8	6 3/8	4 3/8	1/8	6	3/4	6 1/2
RPS-26	3 1/4	5 7/16	1 1/2	1 5/16	12	1 13/16	3/4	7/16	3 7/16	1/4	9	6 1/16	7/8	1/16	3 1/2	4 1/4	1/2	5/16	1/64	5	7 3/8	5 1/8	1/16	6	7/8	7 1/2
RPS-28	3 1/2	5 7/16	1 5/8	1 13/32	12	2 1/16	7/8	7/16	3 7/16	1/4	9	6 1/16	7/8	3/16	3 1/2	4 1/4	1/2	5/16	1/64	5	7 5/8	5 3/8	1/16	6	7/8	7 1/2
RPS-32	4	6 1/4	2	1 3/4	12	2 7/16	1	1/2	3 13/16	5/16	10	6 5/8	1	3/16	4 1/4	5	9/16	3/8	1/64	6	8 13/16	6 1/4	1/8	6	1	8

T. D. indicates tooling dimension. Dimensions in inches.

NOTE.—In column "Size No." R indicates Reverse Gear; P, Propeller Shaft; S, Steel, and the figures the number of eighths of an inch in shaft diameters.

[7]Nuts for Size No. RPS-26 and RPS-28 use 2¼-in. hexagon stock. Nuts for Size No. RPS-32 use 3-in. hexagon stock.

[8]All threads to be S. A. E. pitch, U. S. Form. See pages 4 and 4c, S. A. E. Handbook, Vol. I.

[9]Pilot tolerance, +0.000, −0.001 in. per in. of diameter. Pilot recess tolerance, +0.001, −0.000 in. per in. of diameter.

[10]Figures in column indicate bolt diameters. Use clearance drill of same nominal size for holes B in flange.

controls and has therefore proposed the following recommendation for adoption as S. A. E. Standard:

> For spark and throttle control levers for motorboats intended for one-man control, the spark lever should be shorter than the throttle lever and both should be mounted on stationary sectors. The sectors should be placed so that the levers are moved forward or upward to advance the spark or to open the throttle.
>
> The gearshift lever should be placed so that it is moved forward to pass from neutral to ahead and backward to pass from neutral to reverse.

Shaft Fittings Division

The Shaft Fittings Division has obtained the consensus of opinion of the industry in reference to the present spline fitting standards upon which the following revisions and extensions presented for adoption are based. The revised tolerances and proposed corner radii are recommended as being in accordance with the best engineering practice.

(30) *Four-Spline Fittings*

The Division has recommended that the tolerance for width W of the present S. A. E. Standard for Four-Spline Fittings be increased by 0.001 in., making the total tolerance 0.002 in. for nominal diameters from ¾ to 1¾ in. inclusive and 0.003 in. from 2 to 3 in. inclusive; and that a maximum radius of 0.015 in. be specified for the corners of the splines.

THE DISCUSSION

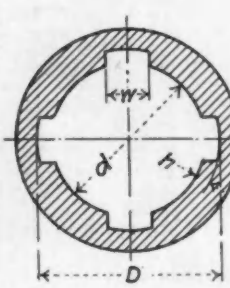
C. W. Spicer:—The increasing use of spline fittings in tractor work as well as some large-size shafts required in war work has made it exceedingly desirable to extend the present standards. It will be recalled that several years ago there was a question whether there should be a specified radius at the corners of the spline. At that time it was considered theoretically desirable but not practically possible. It will be exceedingly difficult and expensive to adopt any given radius, but if the splines are made with square corners these cannot be maintained for any length of time as the corners in the tools and pieces will gradually wear down. It is therefore considered advisable to specify the maximum radius for the corners of the spline. This applies also to the six and ten-Spline Fittings.

(31) *Six-Spline Fittings*

The Division has recommended that the tolerance for width W of the present S. A. E. Standard for Six-Spline Fittings be increased by 0.001 in., making the total tolerance 0.002 in. for nominal diameters from ¾ to 1¾ in. inclusive and 0.003 in. from 2 to 3 in. inclusive; and that a maximum radius of 0.015 in. be specified for the corners of the splines.

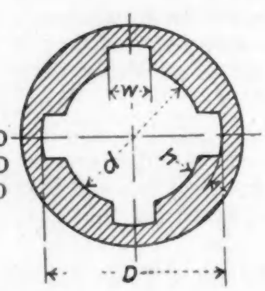
(32) *Sixteen-Spline Fittings*

Twelve-spline fittings in diameters up to and including 5 in. have been used to a limited extent in the automotive industry. The Shaft Fittings Division does not wish to encourage the use of a fitting with the number of splines only slightly more than those of the ten-spline fitting, and has therefore proposed for adoption the following sixteen-spline fitting, which is considered preferable to a twelve-spline fitting. The sixteen-spline fitting will permit greater shaft strength, owing to the



4A
W = 0.241D
h = 0.075D
d = 0.850D

4B
W = 0.241D
h = 0.125D
d = 0.750D



DIMENSIONS FOR FOUR-SPLINE SHAFT FITTINGS

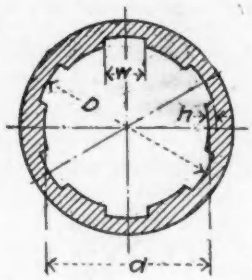
4A—PERMANENT FIT

Nom Diam	D max	D min	d max	d min	W max	W min	h max	h min	[11]T
¾	0.750	0.749	0.637	0.636	0.181	0.179	0.056	0.055	78
⅞	0.875	0.874	0.744	0.743	0.211	0.209	0.066	0.065	107
1	1.000	0.999	0.850	0.849	0.241	0.239	0.075	0.074	139
1⅛	1.125	1.124	0.956	0.955	0.271	0.269	0.084	0.083	175
1¼	1.250	1.249	1.062	1.061	0.301	0.299	0.094	0.093	217
1⅜	1.375	1.374	1.169	1.168	0.331	0.329	0.103	0.102	262
1½	1.500	1.499	1.275	1.274	0.361	0.359	0.112	0.111	311
1⅝	1.625	1.624	1.381	1.380	0.391	0.389	0.122	0.121	367
1¾	1.750	1.749	1.487	1.486	0.422	0.420	0.131	0.130	424
2	2.000	1.998	1.700	1.698	0.482	0.479	0.150	0.148	555
2¼	2.250	2.248	1.912	1.910	0.542	0.539	0.169	0.167	703
2½	2.500	2.498	2.125	2.123	0.602	0.599	0.187	0.185	865
3	3.000	2.998	2.550	2.548	0 723	0.720	0.225	0 223	1,249

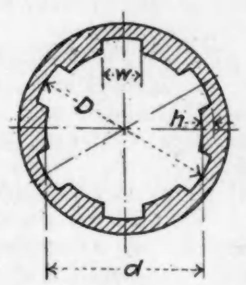
4B—TO SLIDE WHEN NOT UNDER LOAD

Nom Diam	D max	D min	d max	d min	W max	W min	h max	h min	[11]T
¾	0.750	0.749	0.562	0.561	0.181	0.179	0.094	0.093	123
⅞	0.875	0.874	0.656	0.655	0.211	0.209	0.109	0.108	167
1	1.000	0.999	0.750	0.749	0.241	0.239	0.125	0.124	219
1⅛	1.125	1.124	0.844	0.843	0.271	0.269	0.141	0.140	277
1¼	1.250	1.249	0.937	0.936	0.301	0.299	0.156	0.155	341
1⅜	1.375	1.374	1.031	1.030	0.331	0.329	0.172	0.171	414
1½	1.500	1.499	1.125	1.124	0.361	0.359	0.187	0.186	491
1⅝	1.625	1.624	1.219	1.218	0.391	0.389	0.203	0.202	577
1¾	1.750	1.749	1.312	1.311	0.422	0.420	0.219	0.218	670
2	2.000	1.998	1.500	1.498	0.482	0.479	0.250	0.248	875
2¼	2.250	2.248	1.687	1.685	0.542	0.539	0.281	0.279	1,106
2½	2.500	2.498	1.875	1.873	0.602	0.599	0.312	0.310	1,365
3	3.000	2.998	2.250	2.248	0.723	0.720	0.375	0.373	1,969

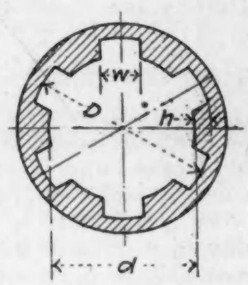
Dimensions in inches.
Radii on corners of splines not to exceed 0.015 in.
Splines shall not be more than 0.006 in. per ft. out of parallel with respect to the axis of the shaft.
[11]Torque=1000x4x mean Rxhx1=pound-inches torque capacity per inch of bearing length at 1000 lb. per sq. in. pressure on sides o splines.
No allowance is made for radii on corners nor for clearances.



6A
W = 0.250D
h = 0.050D
d = 0.900D



6B
W = 0.250D
h = 0.075D
d = 0.850D



6C
W = 0.250D
h = 0.100D
d = 0.800D

DIMENSIONS FOR SIX-SPLINE SHAFT FITTINGS

6A—PERMANENT FIT

Nom Diam	D max	D min	d max	d min	W max	W min	[12]T
3/4	0.750	0.749	0.675	0.674	0.188	0.186	80
7/8	0.875	0.874	0.788	0.787	0.219	0.217	109
1	1.000	0.999	0.900	0.899	0.250	0.248	143
1 1/8	1.125	1.124	1.013	1.012	0.281	0.279	180
1 1/4	1.250	1.249	1.125	1.124	0.313	0.311	223
1 3/8	1.375	1.374	1.238	1.237	0.344	0.342	269
1 1/2	1.500	1.499	1.350	1.349	0.375	0.373	321
1 5/8	1.625	1.624	1.463	1.462	0.406	0.404	376
1 3/4	1.750	1.749	1.575	1.574	0.438	0.436	436
2	2.000	1.998	1.800	1.798	0.500	0.497	570
2 1/4	2.250	2.248	2.025	2.023	0.563	0.560	721
2 1/2	2.500	2.498	2.250	2.248	0.625	0.622	891
3	3.000	2.998	2.700	2.698	0.750	0.747	1,283

6B—TO SLIDE WHEN NOT UNDER LOAD

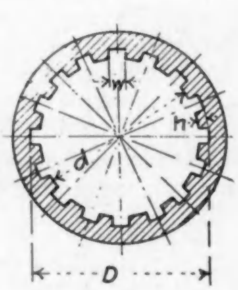
Nom Diam	D max	D min	d max	d min	W max	W min	[12]T
3/4	0.750	0.749	0.638	0.637	0.188	0.186	117
7/8	0.875	0.874	0.744	0.743	0.219	0.217	159
1	1.000	0.999	0.850	0.849	0.250	0.248	208
1 1/8	1.125	1.124	0.956	0.955	0.281	0.279	263
1 1/4	1.250	1.249	1.063	1.062	0.313	0.311	325
1 3/8	1.375	1.374	1.169	1.168	0.344	0.342	393
1 1/2	1.500	1.499	1.275	1.274	0.375	0.373	468
1 5/8	1.625	1.624	1.381	1.380	0.406	0.404	550
1 3/4	1.750	1.749	1.488	1.487	0.438	0.436	637
2	2.000	1.998	1.700	1.698	0.500	0.497	833
2 1/4	2.250	2.248	1.913	1.912	0.563	0.560	1,052
2 1/2	2.500	2.498	2.125	2.123	0.625	0.622	1,300
3	3.000	2.998	2.550	2.548	0.750	0.747	1,873

6C—TO SLIDE WHEN UNDER LOAD

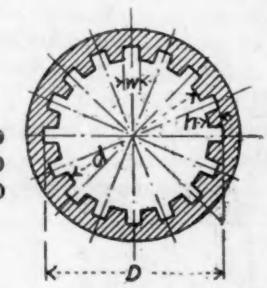
Nom Diam	D max	D min	d max	d min	W max	W min	[12]T
3/4	0.750	0.749	0.600	0.599	0.188	0.186	152
7/8	0.875	0.874	0.700	0.699	0.219	0.217	207
1	1.000	0.999	0.800	0.799	0.250	0.248	270
1 1/8	1.125	1.124	0.900	0.899	0.281	0.279	342
1 1/4	1.250	1.249	1.000	0.999	0.313	0.311	421
1 3/8	1.375	1.374	1.100	1.099	0.344	0.342	510
1 1/2	1.500	1.499	1.200	1.199	0.375	0.373	608
1 5/8	1.625	1.624	1.300	1.299	0.406	0.404	713
1 3/4	1.750	1.749	1.400	1.399	0.438	0.436	827
2	2.000	1.998	1.600	1.598	0.500	0.497	1,080
2 1/4	2.250	2.248	1.800	1.798	0.563	0.560	1,367
2 1/2	2.500	2.498	2.000	1.998	0.625	0.622	1,688
3	3.000	2.998	2.400	2.398	0.750	0.747	2,430

Dimensions in inches.
Radii on corners of splines not to exceed 0.015 in.
Splines shall not be more than 0.006 in. per ft. out of parallel with respect to the axis of the shaft.
[12]Torque = 1000x6x mean Rxhx1 = pound-inches torque capacity per inch of bearing length at 1000 lb. per sq. in. pressure on side of splines
No allowance is made for radii on corners nor for clearances.

16A
W = 0.098D
h = 0.045D
d = 0.910D



16B
W = 0.098D
h = 0.070D
d = 0.860D



16C
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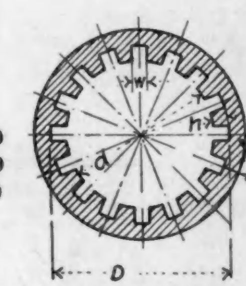
h = 0.095D
d = 0.810D



DIMENSIONS FOR SIXTEEN-SPLINE SHAFT FITTINGS

16A—PERMANENT FIT

Nom Diam	D max	D min	d max	d min	W max	W min	[13]T
2	2.000	1.997	1.820	1.817	0.196	0.193	1,375
2 1/2	2.500	2.497	2.275	2.273	0.245	0.242	2,149
3	3.000	2.997	2.730	2.727	0.294	0.291	3,094
3 1/2	3.500	3.497	3.185	3.182	0.343	0.340	4,212
4	4.000	3.997	3.640	3.637	0.392	0.389	5,501
4 1/2	4.500	4.497	4.095	4.092	0.441	0.438	6,962
5	5.000	4.997	4.550	4.547	0.490	0.487	8,595
5 1/2	5.500	5.497	5.005	5.002	0.539	0.536	10,395
6	6.000	5.997	5.460	5.457	0.588	0.585	12,377

16B—TO SLIDE WHEN NOT UNDER LOAD

Nom Diam	D max	D min	d max	d min	W max	W min	[13]T
2	2.000	1.997	1.720	1.717	0.196	0.193	2,083
2 1/2	2.500	2.497	2.150	2.147	0.245	0.242	3,255
3	3.000	2.997	2.580	2.577	0.294	0.291	4.687
3 1/2	3.500	3.497	3.010	3.007	0.343	0.340	6.378
4	4.000	3.997	3.440	3.437	0.392	0.389	8,333
4 1/2	4.500	4.497	3.870	3.867	0.441	0.438	10,546
5	5.000	4.997	4.300	4.297	0.490	0.487	13,020
5 1/2	5.500	5.497	4.730	4.727	0.539	0.536	15,754
6	6.000	5.997	5.160	5.157	0.588	0.585	18,749

16C—TO SLIDE WHEN UNDER LOAD

Nom Diam	D max	D min	d max	d min	W max	W min	[13]T
2	2.000	1.997	1.620	1.617	0.196	0.193	2,751
2 1/2	2.500	2.497	2.025	2.022	0.245	0.242	4,290
3	3.000	2.997	2.430	2.427	0.294	0.291	6,190
3 1/2	3.500	3.497	2.835	2.832	0.343	0.340	8.426
4	4.000	3.997	3.240	3.237	0.392	0.389	11,005
4 1/2	4.500	4.497	3.645	3.642	0.441	0.438	13,928
5	5.000	4.997	4.050	4.047	0.490	0.487	17,195
5 1/2	5.500	5.497	4.455	4.452	0.539	0.536	20.806
6	6.000	5.997	4.860	4.857	0.588	0.585	24,760

Dimensions in inches.
Radii on corner of splines not to exceed 0.015 in.
Splines shall not be more than 0.006 in. per ft. out of parallel with respect to the axis of the shaft.
[13]Torque = 1000x16x mean Rxhx1 = pound-inches torque capacity per inch of bearing length at 1000 lb. per sq. in. pressure on sides of splines.
No allowance is made for radii on corners nor for clearances.

smaller height of the splines, than either a ten- or twelve-spline fitting.

THE DISCUSSION

C. W. SPICER:—Requirements in war work made it necessary in some cases to use spline fittings as large as 5 in. in diameter. These larger diameters naturally require more than ten splines, and in practice eighteen or twenty splines were used. Twelve splines are also in use but this fitting is so close to the ten-spline that it does not seem desirable to have such a new standard. The sixteen-spline fitting has therefore been developed somewhat as a compromise due to the existing conditions.

(33) *Ten-Spline Fittings*

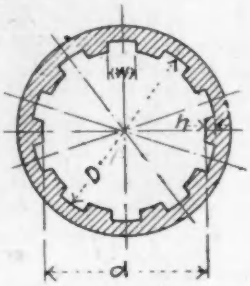
The Division has recommended that the tolerance for width W of the present S. A. E. Standard for Ten-Spline Fittings be increased by 0.001 in., making the total tolerance 0.002 in. for nominal diameters from ¾ to 1¾ in. inclusive and 0.003 in. from 2 to 3 in. inclusive; that a maximum radius of 0.015 in. be specified for the corners of the splines; and that the standard be extended to include 6-in. and intermediate nominal diameters.

STATIONARY ENGINE AND LIGHTING PLANT DIVISION

(34) *Approval of Existing S. A. E. Standards*

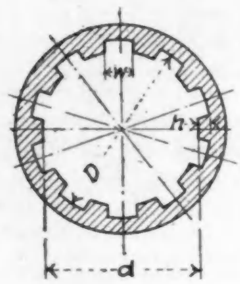
The Division has approved the following S. A. E. Standards and Recommended Practices as satisfactory for stationary engine practice:

	Data Sheet
Adjustable Yoke Rod-Ends	1
Plan Yoke Rod-Ends	1a
Eye Rod-Ends	2
Rod-End Pins	2a
Cotter-Pins	2b
Spark-Plug Shells	3
Nuts for Machine Screws	3c
Screws and Bolts	4, 4a
Screw Threads	4c
Lock Washers	5
Taper Fittings with Castle Nuts	7e
Steel Specifications	9, 9a, 9b, 9c, 9d
Valve Metals	10
Babbitt Metal	11
Bearing Metals	11
Brass Casting Metals	12



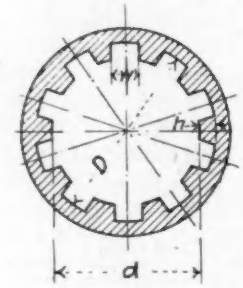
10A

W = 0.156D
h = 0.045D
d = 0.910D



10B

W = 0.156D
h = 0.070D
d = 0.860D



10C

W = 0.156D
h = 0.095D
d = 0.810D

DIMENSIONS FOR TEN-SPLINE SHAFT FITTINGS

10A—PERMANENT FIT								10B—TO SLIDE WHEN NOT UNDER LOAD								10C—TO SLIDE WHEN UNDER LOAD							
Nom Diam	D		d		W		[14]T	Nom Diam	D		d		W		[14]T	Nom Diam	D		d		W		[14]T
	max	min	max	min	max	min			max	min	max	min	max	min			max	min	max	min	max	min	
¾	0.750	0.749	0.683	0.682	0.117	0.115	120	¾	0.750	0.749	0.645	0.644	0.117	0.115	183	¾	0.750	0.749	0.608	0.607	0.117	0.115	241
⅞	0.875	0.874	0.796	0.795	0.137	0.135	165	⅞	0.875	0.874	0.753	0.752	0.137	0.135	248	⅞	0.875	0.874	0.709	0.708	0.137	0.135	329
1	1.000	0.999	0.910	0.909	0.156	0.154	215	1	1.000	0.999	0.860	0.859	0.156	0.154	326	1	1.000	0.999	0.810	0.809	0.156	0.154	430
1⅛	1.125	1.124	1.024	1.023	0.176	0.174	271	1⅛	1.125	1.124	0.968	0.967	0.176	0.174	412	1⅛	1.125	1.124	0.911	0.910	0.176	0.174	545
1¼	1.250	1.249	1.138	1.137	0.195	0.193	336	1¼	1.250	1.249	1.075	1.074	0.195	0.193	508	1¼	1.250	1.249	1.013	1.012	0.195	0.193	672
1⅜	1.375	1.374	1.251	1.250	0.215	0.213	406	1⅜	1.375	1.374	1.183	1.182	0.215	0.213	614	1⅜	1.375	1.374	1.114	1.113	0.215	0.213	813
1½	1.500	1.499	1.365	1.364	0.234	0.232	483	1½	1.500	1.499	1.290	1.289	0.234	0.232	732	1½	1.500	1.499	1.215	1.214	0.234	0.232	967
1⅝	1.625	1.624	1.479	1.478	0.254	0.252	566	1⅝	1.625	1.624	1.398	1.397	0.254	0.252	860	1⅝	1.625	1.624	1.316	1.315	0.254	0.252	1,135
1¾	1.750	1.749	1.593	1.592	0.273	0.271	658	1¾	1.750	1.749	1.505	1.504	0.273	0.271	997	1¾	1.750	1.749	1.418	1.417	0.273	0.271	1,316
2	2.000	1.998	1.820	1.818	0.312	0.309	860	2	2.000	1.998	1.720	1.718	0.312	0.309	1.302	2	2.000	1.998	1.620	1.618	0.312	0.309	1.720
2¼	2.250	2.248	2.048	2.046	0.351	0.348	1.088	2¼	2.250	2.248	1.935	1.933	0.351	0.348	1.647	2¼	2.250	2.248	1.823	1.821	0.351	0.348	2,176
2½	2.500	2.498	2.275	2.273	0.390	0.387	1,343	2½	2.500	2.498	2.150	2.148	0.390	0.387	2,034	2½	2.500	2.498	2.025	2.023	0.390	0.387	2,688
3	3.000	2.998	2.730	2.728	0.468	0.465	1,934	3	3.000	2.998	2.580	2.578	0.468	0.465	2,929	3	3.000	2.998	2.430	2.428	0.468	0.465	3,880
3½	3.500	3.497	3.185	3.182	0.546	0.543	2,632	3½	3.500	3.497	3.010	3.007	0.546	0.543	3,987	3½	3.500	3.497	2.835	2.832	0.546	0.543	5,266
4	4.000	3.997	3.640	3.637	0.624	0.621	3,438	4	4.000	3.997	3.440	3.437	0.624	0.621	5.208	4	4.000	3.997	3.240	3.237	0.624	0.621	6,878
4½	4.500	4.497	4.095	4.092	0.702	0.699	4,351	4½	4.500	4.497	3.870	3.867	0.702	0.699	6.591	4½	4.500	4.497	3.645	3.642	0.702	0.699	8.705
5	5.000	4.997	4.550	4.547	0.780	0.777	5.371	5	5.000	4.997	4.300	4.297	0.780	0.777	8.137	5	5.000	4.997	4.050	4.047	0.780	0.777	10.746
5½	5.500	5.497	5.005	5.002	0.858	0.855	6.500	5½	5.500	5.497	4.730	4.727	0.858	0.855	9,846	5½	5.500	5.497	4.455	4.452	0.858	0.855	13.003
6	6.000	5.997	5.460	5.457	0.936	0.933	7,735	6	6.000	5.997	5.160	5.157	0.936	0.933	11.718	6	6.000	5.997	4.860	4.857	0.936	0.933	15.475

Dimensions in inches.
Radii on corner of splines not to exceed 0.015 in.
Splines shall not be more than 0.006 in. per ft. out of parallel with respect to the axis of the shaft.
[14]Torque=1000x10x mean Rxhx1=pound-inches torque capacity per inch of bearing length at 1000 lb. per sq. in. pressure on sides of splines.
No allowance is made for radii on corners nor for clearances.

	Data Sheet
Cast Manganese Bronze	12a
Manganese Bronze Sheets and Rods	12b, 12c
Hard Cast Bronze	12d
Gear Bronze	12d
Aluminum Alloys	13
Brass Sheets and Strips	13a
Brass Rods	13b
Tobin Bronze Rods	13b
Non-Ferrous Metal Tubing	13c
Pitch of Silent Chains	14b
Widths of Silent Chains	14b
Roller-Chain Dimensions	14c
Round Tension Test-Specimen	15
Flat Tension Test-Specimen	15xa
Flat Tension Test-Specimen	15a
Shock Test-Specimen	15xb
Gray-Iron Test-Specimen	15b, 15c
Brinell Hardness Test	15xd
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Annular Ball Bearings	29, 29a, 29b, 29c
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Magneto Dimensions	36, 36xa
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Dimensions of Lead Batteries for Lighting and for Combined Lighting and Starting Service	48c
Temperature Test of Insulating Materials of Electrical Apparatus on Gasoline Automobiles	48g
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Round Pipe Flanges	57a

(35) *Voltage and Capacity Ratings, Isolated Electric Lighting Plants*

The Stationary Engine and Lighting Plant Division recommended that the nominal voltage of 32 for the 5-kw. normal generator rating in the present S. A. E. Recommended Practice for Voltage and Capacity Ratings of Isolated Electric Lighting Plants, page 57, S. A. E. Handbook, Vol. I, should be omitted. The Division does not believe that the 32-volt rating should be omitted from the 3-kw. size, as it is used frequently for motion-picture operation.

VOLTAGE AND CAPACITY RATINGS

Normal Generator Rating, kw.	[15]Nominal Voltage	Engine and Generator Speed, r.p.m.
½, ¾, 1, 1½, 2, 3	32 or 110	1,200 or 1,800
5	110	1,200 or 1,800

[15] Sixteen cells for 32-volt and fifty-six cells for 110-volt lead batteries.

(36) *Cast-Iron Carbureter Flanges*

The Division recommended for adoption as S. A. E. Recommended Practice that flanges for cast-iron carbureters be specified as the next larger size for each nominal diameter of opening given in the S. A. E. Standard for Carbureter Flanges, pages 35 and 35xa, S. A. E. Handbook, Vol. I.

Tire and Rim Division

(37) *Pneumatic Tires for Passenger Cars and Commercial Vehicles*

The Tire and Rim Division has recommended that the 33x4½ and the 34x5-in. pneumatic tire sizes be included in the list of S. A. E. Standard regular sizes to meet the growing demand for 24-in. wheels with 4½ and 5-in. tires.

At the meeting of the Joint Executive Committee of the Pneumatic and Solid Tire Divisions of the Rubber Association of America on Nov. 26, 1919, the insertion of the 34 x 5-in. size was approved. At the time of writing this report the 33 x 4½ and 42 x 9-in. sizes are scheduled for consideration and approval at the next meeting of the Committee of the Rubber Association.

The Executive Committee of the Tire and Rim Association at its meeting on Dec. 10, 1919, approved the addition of the 33 x 4½ and 34 x 5-in. sizes and agreed upon the dimensions for the 42 x 9-in. rim.

The addition to the S. A. E. Standard of the 44 x 10-in. size will be considered by the 1920 Tire and Rim Division of the Standards Committee to complete the list of regular and oversize tires in the Standard lineup. The present S. A. E. Recommended Practice for the carrying capacity and inflation pressure of the 9 and 10-in. tires is printed

on page 8h, S. A. E. Handbook, Vol. I. All sizes are for straight-side rims.

Nominal Tire and Rim Size		Oversize Tire		Tire Seat Diam. (Rim)	
In.	Mm.	In.	Mm.	In.	Mm.
33x4½	120/610	34x5	135/610	24	610
34x5	135/610	36x6	150/610	24	610
*40x8	200/610	42x9	225/610	24	610
42x9	225/610	.....	....	24	610

*The 40x8-in. tire is already standard for regular equipment, but is submitted for approval of the 42x9-in. tire as oversize equipment.

(38) *Pneumatic Tires for Motorcycles*

The recommended practice for pneumatic tire sizes adopted by the Society two years ago was 2¼ and 2½ in. for the BB rim section and 2¾ and 3 in. for the CC rim section. As a result of the experience of the Motor Transport Corps here and in France with motorcycle transportation, and owing to the tendency in commercial motorcycle practice being toward heavyweight machines the Tire and Rim Division has carefully reviewed the subject. The motorcycle and tire manufacturers were canvassed to obtain the necessary data for suitable revision of the recommended practice. General opinion among motorcycle manufacturers is that tires should not be smaller than 3 in. as tires under this size have neither the capacity nor wearing qualities necessary for the heavy machines built to-day. The recommendation of a 29 x 3½-in. tire has been considered, but the main objection from the motorcycle manufacturers seems to be that this size tire interferes with the forks and mudguards of the machine unless extensive changes are made in the motorcycle design, to which they are unfavorably inclined.

The members of the Division believe that a definite schedule of motorcycle tire sizes should be adopted, and have recommended the following tire sizes for adoption as S. A. E. Recommended Practice, all sizes to be used on the CC rim section:

Tire Size		Maximum Load		Corresponding Inflation Pressure	
In.	Mm.	Lb. per Tire	Kg. per Tire	Lb. per Sq. In.	Kg. per Sq. Cm.
26x3	75/510	325	147.5	40	2.81
27x3½	90/510	400	181.4	45	3.16
28x3	75/560	325	147.5	40	2.81

The maximum loads and corresponding inflation pressures are already S. A. E. Standard and are printed on page 8h, S. A. E. Handbook, Vol. I.

The present recommended practice for the 28x3 wheel rim and lacing of spokes is printed on page 8-l, S. A. E. Handbook, Vol. I.

(39) *Solid Tire Sizes*

With the advent of trailers and semi-trailers as an important factor in the automotive industry, there is also a demand for adequate tire standards for this type of vehicle. The Tire and Rim Division is of the opinion that it is unnecessary to have a separate standard for trailer tires as the present standard for pneumatic tires is considered adequate, but it is considered desirable and the Division has recommended that the following solid tire sizes be included in the present S. A. E. Standard for Solid Tire Sizes, page 8bb, S. A. E. Handbook, Vol. I.

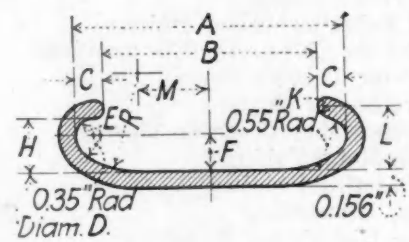
The Solid Tire Manufacturers' Division of the Rubber Association of America voted to incorporate the proposed sizes in the list of standard solid tire sizes at the meeting on Sept. 24, 1919.

SOLID TIRE SIZES

In.	Mm.
36x3	75/762
34x6	150/711
34x7	175/711

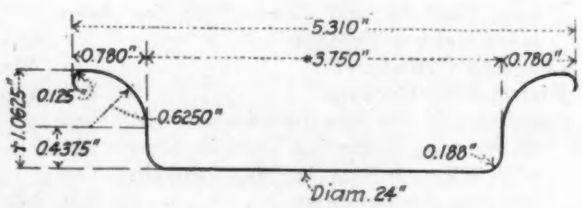
(40) *Rim Sections and Contours for Pneumatic Tires (Correction)*

In the cut for the clincher rim section, page 8ha, S. A. E. Handbook, Vol. I, the 0.55-in. radius is indicated as being to the outside of the rim, whereas it should be to the inside of the rim. It is therefore recommended by the Division that the present S. A. E. Standard for Rim Sections and Contours for Pneumatic Tires be corrected accordingly. This corresponds with the corrected Tire and Rim Association record.



(41) *Rim Sections and Contours for Pneumatic Tires (Extension)*

The present S. A. E. Standard, pages 8ha and 8hb, S. A. E. Handbook, Vol. I, does not include a 5-in. rim section. The dimensions given in the accompanying illustration have been worked out and adopted by the Tire and Rim Association and are presented above for adoption as S. A. E. Standard.



Nominal tire and rim size: 34 x 5 in.
Type of rim: straight-side.
Rim diameter: 24 in.
Rim circumference: 75.398 ± 0.047 in.
*Tolerances: plus or minus 0.047 in.
†Tolerances: plus 0.031 in., minus 0.000.

5-In. Pneumatic Tire Rim Contour

(42) *Solid Tire Sections*

It is recommended by the Tire and Rim Division that the present S. A. E. Standard for Solid Tire Sections be completed by inserting the area of 5 sq. in. for the

3-in. tire size. A canvass of the entire industry indicated almost unanimous approval of this area for the 3-in. size. Since the sectional rubber area should be expressed either with an allowable tolerance or as a minimum it is the consensus of opinion that the column in the printed standard should have the heading, "Minimum Total Sectional Area of Rubber, Sq. In."

(43) *Bolt Equipment for Solid-Tire Side Flanges*

The Tire and Rim Division recommends that the present S. A. E. Standard for Bolt Equipment for Solid-Tire Side Flanges, page 8xb, S. A. E. Handbook, Vol. I, be revised to include the 32, 34, 36 and 40-in. tire sizes only, thus eliminating the non-standard 30, 38 and 42-in. solid tire sizes.

(44) *Wood Spoke Dimensions for Commercial Vehicle Wheels*

The Tire and Rim Division has recommended that dimension *D* in the illustration for wood-spoke dimensions for commercial vehicle wheels, page 8-o, S. A. E. Handbook, Vol. I, and letter *D* in the heading for the column for the nominal spoke size be omitted. The dimensions for *D* (unfinished thickness of spoke between flanges) are not pertinent to the present standard.

Tractor Division

(45) *Tractor Belt Speeds*

One of the first tractor standards adopted by the Society over two years ago was the belt-speed standard of 2600 ft. per min. As there was little uniformity of practice at that time, it was felt that not much more could be accomplished. Over a year ago there was evidence of a strong desire for adequate standardization owing to rapid development of the tractor industry and the relation between factors affecting design, marketing, operation and maintenance common to both agricultural implements and tractors. In February, 1919, a Sub-Division of the Tractor Division was appointed which has made a very careful and detailed canvass of the tractor, stationary engine and implement manufacturers to ascertain present practice and to bring the industries into agreement on an adequate and acceptable standard.

It was obvious that the most desirable solution of the problem would be one standard belt speed, but it was also obvious that this would not be practical or satisfactory for a number of reasons. A large thresher having a cylinder speed of 1200 r.p.m. must have a drive pulley of 10 to 12 in. in diameter to have sufficient belt contact to transmit the required power. Consequently a belt speed of 3400 to 3500 ft. per min. is necessary for this class of equipment. A small thresher with cylinder speeds reduced to 800 or 900 r.p.m. for threshing rice or to avoid cracking grain or similar products could not be equipped with a drive pulley large enough for this high belt speed on account of the widening out of the separator directly to the rear of the cylinder. Many power-operated farm implements such as feed grinders, ensilage cutters, shredders, small and medium-sized engines, etc., cannot for many reasons be equipped with pulleys large enough for this high belt speed.

The intention of the Sub-Division was originally to propose two or possibly three belt speeds, but a careful analysis of the mass of data obtained warranted the conclusion that it would not be possible or practical to propose less than four belt speeds. In present practice there are from ten to fifteen different belt speeds being used by tractor and agricultural implement manufacturers and it is evident that a reduction of this number of belt speeds to a standard of four speeds is very desirable.

The proposed plan of standardization does not contemplate four standards for every size of tractor, but it is believed will be applicable somewhat as follows:

(1) A belt speed of 1500 ft. per min. is the present standard for stationary engine work, and if used at all on tractors would be confined to the very smallest sizes
(2) Belt speeds of 2000 and 2600 ft. per min. would be applicable on tractors in the two-plow or 8-16, 9-18 and 10-20-hp. classes
(3) Belt speeds of 2600 and 3000 ft. per min. would be applicable on tractors of the 10-20, 12-25 and 15-27-hp. and three-plow classes
(4) Belt speeds of 3000 and 3500 ft. per min. would be applicable on tractors of the largest sizes

It is believed that the proposed plan of standardization will reduce the pulley requirements for power-driven farm equipment to not more than two or perhaps three different sizes of pulleys for each machine and to possibly not more than two belt speeds for each size of tractor.

E. A. Johnston, chairman of the Sub-Division on Belt Speeds, reported on Oct. 31 substantially as follows:

REPORT OF SUB-DIVISION ON BELT SPEEDS

In the endeavor to arrive at a series of standard belt speeds for power-driven farm machinery a very thorough analysis of available information was made by the Sub-Division. A brief summary of the procedure will be of value in considering the recommendations proposed for adoption.

Machinery Considered The farm power-driven machine groups considered were:

Tractors	Corn Shellers
Threshers	Corn Huskers
Feed Grinders	Hay Presses
Ensilage Cutters	Stationary Engines (small)

Prevailing Belt Speeds Each of the above groups was analyzed in a number of ways. The controlling factors for the standard and special pulley sizes and speeds for these machines are:

Consideration of space, power and revolutions per minute required for clutch and plain pulleys for both power-driven and power-driving equipment with respect to design.

Meeting speed requirements when receiving power from a variety of power-driving equipment, for both stationary engines and tractors, by clutch and plain pulleys.

Meeting speed requirements of the product operated upon by the power-driven equipment.

Consideration of Complexity Consideration was given to the difficulties involved in the adoption of standard belt speeds. It is evident that it is more expensive to meet changes in clutch and plain pulley sizes controlling belt speeds in the tractor group than in any other group. In the thresher group changes involve slippage and interference more than in any other group.

Simple belt speed or pulley changes can be more readily and less expensively made in the last five groups. Due emphasis has therefore been given this phase in selecting the limited number of belt speeds recommended. The experience and volume of output of the various makers of equipment has also been duly weighed. With these ideas in view each group of machines was reviewed by plotting belt speeds against horsepower, belt speeds against pulley sizes, pulley sizes and widths against horsepower transmitted, and belt speeds against machine sizes. From this tabulated data the accompanying chart on belt speeds was made.

In interpreting the chart, line one shows the four recommended speeds of 1500, 2600, 3000 and 3500 ft. per min. As engines do not always run at the exact speed and as power-driven machines will operate well at reasonable variations from these definite speeds, as shown by questionnaire answers, lines two and three were plotted with 5 and 7 per cent variations above and below the recommended speeds. It will be noted that the three higher belt speeds thus form an almost continuous series of speeds without overlapping.

Lines four to ten represent the massed speeds obtaining in present practice, thus eliminating the scattered speeds used as the result of inexperience, unsuitable designs or special requirements. It will

BELT SPEED IN FEET PER MINUTE

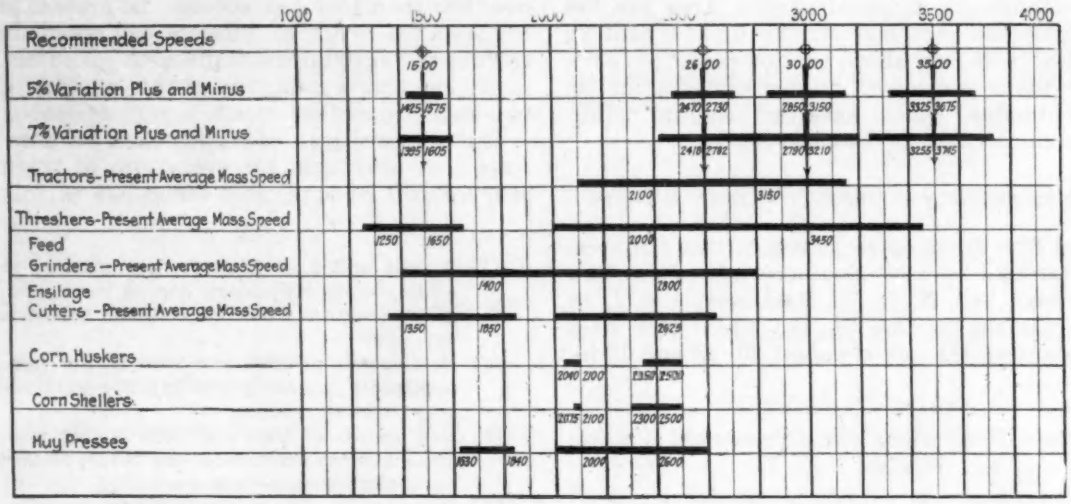


CHART SHOWING PRESENT BELT SPEED PRACTICE

be seen that mass speeds in feet per minute of the several groups of machines range as follows:

Tractors	2100 to 3150
Threshers	1250 to 1650, 2000 to 3450
Feed Grinders	1400 to 2800
Ensilage Cutters	1300 to 1850, 2000 to 2625
Corn Huskers	2040 to 2100, 2350 to 2500
Corn Shellers	2075 to 2100, 2300 to 2500
Hay Presses	1630 to 1840, 2000 to 2600

There might be some question as to why 2000 ft. per min. was not included in the recommendation, especially since there are so many machines operating at or near this speed. From the chart it will be seen that 2000 ft. per min. is the lower limit for the higher series of speeds, that the average is about 2400 ft. per min., but that tractors and threshers average about 2600 ft. per min. It seemed advisable therefore to select 2600 instead of 2400 ft. per min.

There were 40 replies to the 110 questionnaires sent out or 36.[illegible] per cent.

The replies were quite unanimous in approving the speeds selected as shown by the following percentages, counting only those who gave positive answers or indicated that they would adopt the speeds recommended.

Speed. ft. per min.	Total Number of Replies	Replies Approving of Speeds	Per Cent
1,500	39	37	97.5
2,600	40	39	97.5
3,000	40	39	97.5
3,500	40	38	95.0

Those suggesting additional speeds were 6 out of the 40 or 15 per cent, as follows:

One for 3250 ft. per min. for threshers
Two for 3200 ft. per min. for tractors and ensilage cutters
One for 2000 ft. per min. to meet present practice and ensilage cutters
One for 2000 ft. per min. because range of 1500 to 2600 ft. per min. is too great
One for slow speeds for pumps and cream separators

As to which speeds could be left out, the answers were rather from the view of their individual product.

Speed to be Omitted ft. per min.	Total Number of Replies	Replies in Favor of Omission	Per Cent
1,500	40	5	12.5
2,600	40	1	2.5
3,000	40	2	5.0
3,500	40	7	17.5

As to the probable reduction possible in the number of stock pulley sizes, as a result of standardizing on the proposed belt speeds, the answers indicated the following:

Present Number of Pulley Sizes	Probable Number of Pulley Sizes	Reduction in Per Cent
15	3	80
30	15	50
10	4	60
28	9	68
14	1	93
15	9	40
333	133	40
55	15	73

Belt widths recommended were

4 in. for small ensilage cutters
6 in. for feed grinders and ensilage cutters
7 in. for ensilage cutters
8 in. for ensilage cutters, feed grinders, hay presses and corn shellers

The replies were quite unanimous in approving the series of belt widths proposed, 28 out of 29 or 96.5 per cent being in favor of adopting them.

As to the possible variation allowable in the belt speed, the replies varied in accordance with the type of machine and the product. Replies indicated that from minus 10 to plus 30 per cent of their rated speeds were allowable variations. The mass data varied between minus 5 and plus 15 per cent. This supported the assumption of 7 per cent plus or minus allowable variation in determining the belt speeds.

Replies unanimously favored the adoption of a series of standard speeds.

In reply to the question regarding any special conditions that could not be covered by the proposed belt speeds, there was but one answer dealing with very low speeds in the low-powered machines, which are really outside the transmitted horsepower considered in this report.

Conclusion It is therefore recommended by the Sub-Division

That a minimum series of belt speeds for farm power-driving and power-driven machines be adopted as standard.

That, owing to the variety of requirements and the present wide range of belt speeds, future designs of machines and equipment be made to conform to the recommended minimum number of belt speeds.

That in so far as possible present farm power machines be regularly equipped with pulleys to meet these standard speeds.

That 1500, 2600, 3000 and 3500 ft. per min. be regularly adopted as standard belt speeds covering the principal line of farm power-driving and power-driven machines of 10 hp. and over.

That the general large reduction in repair stocks, obsolete sizes required of manufacturers and the smaller variety of replacement parts required by users will effect a considerable saving to all concerned.

That the adoption of the proposed belt speeds need not cause any given manufacturer to discard speeds and pulley sizes either desirable or necessary to his trade but that such cases be considered as special.

That standard belt widths of 4, 6, 7 and 8 in. be adopted.

On Nov. 7 there was a meeting in the offices of the National Implement and Vehicle Association at which there were representatives of the Tractor and Thresher Division of that Association, the Society of Agricultural Engineers, E. A. Johnston, chairman of the S. A. E. Tractor Sub-Division on Belt Speeds and other unofficial S. A. E. representatives. Mr. Johnston's report on belt speeds was discussed and a resolution voted unanimously to recommend to their respective Societies that the belt speeds proposed by Mr. Johnston be adopted as standard.

However, upon later recommendation of the Tractor and Thresher Division of the National Implement and Vehicle Association, belt speeds of 2000 and 3250 ft. per

min. were added to the proposal as being favored by the tractor manufacturers in that Association. The extended proposal has been submitted to the members of the Association interested in power equipment, but no action will be taken by that organization until the speeds are adopted by the Society or the American Society of Agricultural Engineers.

If it seems advisable later on to add another belt speed or otherwise revise the standard, the subject can be presented again to the Standards Committee and the Society for consideration.

The Tractor Division has therefore recommended for adoption as S. A. E. Standard belt speeds of 1500, 2600, 3000 and 3500 ft. per min. for farm power-driving and power-driven machines of 10 hp. and over as reported by the Sub-Division.

(46) *Tractor Belt and Pulley Widths*

The tractor belt and pulley width standard adopted by the Society in August, 1918, was not completed at that time as there was uncertainty as to what belt width was most desirable for transmitting over 30 hp. As belt widths are an important factor in determining belt speeds the report of the Sub-Division includes belt widths, and the Tractor Division has recommended that the present S. A. E. Standard for belt widths be revised to conform to the following proposal:

TRACTOR BELT AND PULLEY WIDTHS

Horsepower	Pulley Width, in.	Maximum Belt Widths, in.
10 to 20	4½ or 6½	4 or 6
20 to 30	7½	7
Above 30	8½	8

Truck Standards Division

(47) *Four-Wheel Trailer Hitches*

This subject was suggested for standardization by trailer manufacturers who appreciate the economic value of having all trucks and trailers interchangeable. Data, obtained from truck and trailer manufacturers showing present trailer hitch practice, were reviewed by a Sub-Division acting in cooperation with a special committee appointed by the Trailer Manufacturers Association and a tentative report was made. The report was then sent to truck and trailer manufacturers and their comments obtained. These together with the report were reviewed by the Truck Standards Division and as the comments indicated that the U. S. Government pintle hook was used to a large extent the report was revised to permit its use.

It is not considered desirable to give a fixed dimension from the axis of the drawbar head to the ground because present truck frame heights are approximately the same within required limits. Lost motion between the eye and the drawbar head should be kept as small as possible to prevent the swaying of trailers in operation.

The Truck Standards Division has therefore recommended for adoption as S. A. E. Recommended Practice the following proposal for Four-Wheel Trailer Hitches.

A spring, mounted preferably in the drawbar head,[16] shall be used in the trailer hitch and may be either exposed or enclosed as desired by the manufacturer.

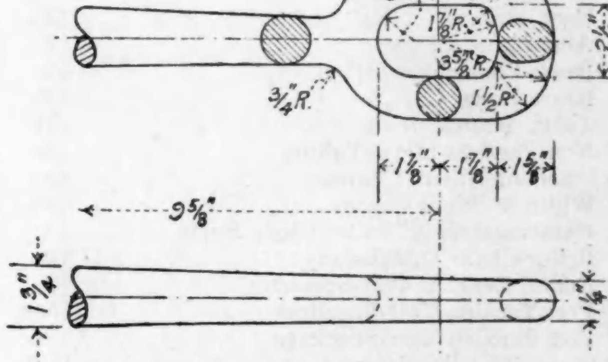
Provision shall be made for chains or other safety devices in addition to the coupling link.[17] A ⅝-in. chain eye clevis with a ¾-in. bolt shall be provided for attaching a safety chain, the clevis to be located directly under the axis of the drawbar head or as near to this position as possible. If placed above the axis it is likely to interfere with operation of the drawbar head.

The axis of the drawbar head shall be located both vertically and horizontally approximately in the center of the frame of the chassis.

The eye[18] for the coupling link shall conform with the dimensions given in the accompanying drawing.

The drawbar-head hook shall be made so as to take the recommended size of eye.

The coupling link shall carry the eye, but when no coupling link is used the eye shall be mounted on the trailer head.[19] The eye shall be free to rotate so that it



DIMENSIONS FOR PROPOSED TRAILER EYE.

can be placed in either a horizontal or vertical position. When the drawbar head can be rotated the eye on the coupling link or trailer head shall be locked so as to prevent it from rotating.

[The name of this subject was changed from Trailer Hitches to that printed above].

(48) *Approval of Existing Standards*

The Truck Standards Division has reviewed the present S. A. E. Standards and Recommended Practices and recommends the following as suitable for commercial vehicle practice. Other present S. A. E. Standards and Recommended Practices are referred to their respective Divisions for modification or extension to adequately answer requirements.

	Data Sheet
Adjustable Yoke Rods-Ends	1
Plain Yoke Rod-Ends	1a
Eye Rod-Ends	2
Rod-End Pins	2a
Cotter Pins	2b
Spark-Plug Shells	3
Nuts for Machine Screws	3c
Screws and Bolts	4, 4a
Screw Threads	4c
Lock Washers	5
Square Broached Fittings	7
6-Spline Fittings	7b
10-Spline Fittings	7c
4-Spline Fittings	7d
Taper Fittings with Castle Nuts	7e
Solid Tire and Wheel Diameters, Wheel Circumferences	8
Measuring of Solid Tire Widths	8

[16] That part which is on the truck.
[17] That part which is between the truck and the trailer.
[18] This size of eye will take the U. S. Government pintle hook.
[19] That part which is on the trailer.

	Data Sheet
Leaf Point Nomenclature	49a
Finish of Springs	49a
Leaf Points	49a
Center Bolts	49b
Rebound Clips, Spacers and Bolts	49c
Frame Brackets	49e
Offset and Resulting Ends	49f
Leaf Spring Specifications	49i to 49t

Attendance at Meeting

The members of the Standards Committee and the Society and the guests in attendance were:

Standards Committee Members

Alborn, F. G.
Andrew, F. W.
Aull, J. J.
Bachman, B. B.
Baker, W. C.
Batt, W. L.
Bauder, Paul F.
Bijur, Joseph
Black, Archibald
Blair, B. H.
Bott, George R.
Bradley, C. I.
Brautigan, H. H.
Broege, R. J.
Brumbaugh, A. K.
Burnett, Robert S.
Carlton, C. C.
Chambers, D. F.
Chryst, W. A.
Clarkson, C. F.
Coleman, J. R.
Copland, A. W.
Crane, H. M.
Darrow, Burgess
Diefendorf, W. H.
Diffin, F. G.
Dudley, A. M.
Ehrman, E. H.
Franklin, C. B.
Fretz, E. S.
Gilchrist, C. F.
Gilligan, F. P.
Goldsmith, F. C.
Graham, Walter F.
Gunn, E. G.
Hale, J. E.
Hargraves, A.
Harley, W. S.
Hecox, Major F. C.
Heldt, P. M.
Hess, S. P.
Heywood, Charles E.
Hinkley, C. C.
Holt, J. W.
Hulse, E. G.
Ickes, E. T.
Kalb, L. P.
Keilholtz, L. S.
Keys, W. C.
Knowles, W. H.
Libby, A. D. T.
McCleary, F. E.
McKay, W. A.
McMahon, Harry R.
MacGregor, James S.
Manly, C. M.
Michel, C. A.
Newkirk, W. M.
Norris, G. L.
Norton, W. T., Jr.
Peterson, W. C.
Pierce, H. S.
Sharp, Dr. C. H.
Snow, H. C.
Spicer, C. W.
Stagg, H. J.
Strickland, W. R.
Sweet, E. E.
Thacher, S. P.
Townsend, R. A.
Tubbs, George E.
Tuttle, J. C.
Van Blerck, Joseph
Wall, W. G.
Wells, R. E.
Wood, H. F.

Society Members and Guests

Almquist, K.
Bailey, Charles A.
Bailey, E. W. M.
Bass, M. R.
Baumgartner, W. J.
Bell, John T. R.
Benner, H. W.
Braender, F. J.
Breeze, A. B.
Brehmer, William
Brunner, H. E.
Buckingham, E.
Busby, E. B.
Call, C. A.
Case, F. P.
Clark, E. L.
Clark, Lieut.-Col. V. E.
Coleman, T. E.
Crawford, J. T.
Cunningham, Dr. R. H.
Davies, D. P.
Denyes, H. M.
Deyo, A. K.
Diamant, N. S.
Dickinson, Dr. H. C.
Duckworth, C. H.
Edwards, I. W.
Figgie, H. E.
Flynn, Gregory
Frehse, A. W.
Gage, V. R.
Gillmer, G. W.
Godley, C. E.
Gunster, E. H.
Halbleib, J. C.
Harleman, S. T.
Harrison, O. L.
Hawke, C. E.
Heath, L. J.
Hoopes, Russell
Horner, L. S.
Howell, J. Bertram
Hutchins, W. H.
Jackson, H. W.
Jaschka, John H.
Johnson, C. R.
Jominy, W. E.
Jones, J. E.
Kaiser, William L.
Kilborn, K. B.
Kliesrath, V. W.
Kononoff, B. A.
Kressler, O.
Lavery, George L.
Laycock, A. M.
Lutes, H.
Lyon, C. L.
McDonald, A. J.
McIntyre, M. M.
MacDonald, H. P.
MacFarland, A. T.
MacKenzie, D.
Margolin, G.
Moskovics, F. E.
Myers, Cornelius T.
Myers, J. L.
Newbold, R. M.
Nielsen, V. A.
Nogle, Byron S.
Northrup, L. M.
Pack, Charles
Pomeroy, T. M.
Quinn, C. J.
Rheinfrank, Eugene
Richards, J. D. F.
Royce, A. C.
Rugg, H. M.
Sargent, G. W.
Schatz, H. A.
Sells, O. P.
Shaffer, R. A.
Shailer, Robert A.
Sivyer, F. L.
Slack, F. W.
Slocum, Frank S.
Smith, F. E.
Sparrow, S. W.
Stern, M.
Swift, William K.
Taub, Alex.
Tavener, C. H., Jr.
Turner, Albert
Udy, I. V.
Veal, C. B.
Verity, C. W.
Ward, L. M.
Watkin, L. M., Jr.
White, H. S.
Williams, L. W.
Woodward, R. W.
Young, Conrad H.
Young, Capt. G. R.
Young, O. W.

BENZOL RECOVERY IN ENGLAND

It has often been argued that owing to the ubiquity of gasworks there should be no difficulty in arranging for the registry of all such works as depots for supply of benzol. In this respect, however, it is usually forgotten that benzol recovery is scarcely a profitable operation for the smaller undertakings, and as regards the larger towns it must be remembered that 90 per cent of the gasworks removing the spirit obtain it in the crude form only. The crude spirit is, consequently, dispatched to other centers, where apparatus for the somewhat complicated process of rectification is available. Thus, supply depots are restricted to the comparatively small number of rectification centers. The handling necessitated is, moreover, a costly item. The day may, of course, come when each gasworks is provided with its own refining apparatus, but hitherto this process has been considered profitable only if carried out on a large scale, although diminutive apparatus for the purpose has now been introduced and is, so far as can be gathered, on order for a number of medium-sized gasworks.

As regards distribution, more particularly in the convenient 2-gal. can, there is no doubt that the whole problem would be greatly simplified if use could be made of the existing gasoline organization for the purpose. It is understood that the gasoline interests have been approached in the matter, and that the benzol producers consider a charge of approximately 2d. per gal. as appropriate for this concession. It can, however, be easily understood that the gasoline purveyors, being in competition with the home-produced spirit, scarcely look at the matter from the same point of view, and it is believed that they demand as much as 5d. per gal. for affording the necessary facilities. Accordingly, as the benzol producers are, when properly organized, in a sufficiently strong position to take up an independent line, there seems every possibility that the motorist must content himself for some few months with the present arrangements until a separate and effective distribution plan is set in motion.

Small Rectification Plants

The erection of rectification apparatus at all gasworks where the recovery of the crude spirit is carried out would immensely simplify the distribution problem. It is for this reason that as much publicity as possible should be given to the fact that a comparatively simple plant, suitable for works producing no more than 50 gal. of crude spirit per day, has now been introduced for the purpose. This means that gasworks with a manufacturing capacity down to a limit of about 100,000,000 cu. ft. of gas per annum could be rated as suppliers. Spread over England, exclusive of Wales and Scotland, there are approximately 280 gas undertakings of this size and upward, apart from the numerous coke-oven establishments, which are nearly all located in the north. Accordingly, if so many centers could be established, nearly every district would be provided for, while the cost of transport would be reduced to a trifling figure.

As regards the new motor-spirit plants for use in the smaller works, that operated on the Wilton principle appears to combine simplicity with a moderate capital outlay. The crude benzol, recovered in the ordinary manner by the com-

paratively simple process of oil-washing, is in the first instance run into a crude still, which consists merely of a small boiler to which is attached a simple column. Distillation is effected by steam coils, and the light products are distilled off, passing through a water-cooled condenser of ordinary construction, from which, in the liquid state, they gravitate to the washer. In the larger motor-spirit plants it is usual to arrange for secondary distillation—that is to say, the once-run spirit is collected in a separate storage tank and subjected to further distillation in the secondary still before undergoing the washing process. This secondary distillation, however, may be eliminated if the precaution is taken to work up the original crude spirit to a strength of from 70 to 80 per cent at 120 deg. cent.

One of the contingent advantages attached to the process of recovering crude benzol is the freedom gained from naphthalene troubles. The wash-oil with which the coal gas is treated, if of the correct composition, not only removes the light hydrocarbons but the greater proportion of the naphthalene as well. Benzol recovery thus offers a further very substantial inducement. It is for this reason that the residue which remains in the crude still of the motor-spirit plant consists of creosote oil highly charged with naphthalene. The economics of the process are such, however, that no product need now be regarded as waste, so that the creosote oil may be separated and sold as a by-product, while the naphthalene may also be refined and will then command a good price. The creosote, of course, is derived from the original wash-oil employed, a small proportion of which invariably comes over with the crude spirit during primary distillation.

As regards the light products obtained from the still of the motor-spirit plant, these contain such impurities as tar acids and tar bases and sulphur compounds. These must be removed, and for that purpose the distillate is treated with sulphuric acid and caustic soda. In the larger rectification plants agitation during washing is carried out by mechanical means, but with the plant for the small gasworks hand agitation is arranged for to avoid every possible complication. After the purification treatment the distillates are run by gravitation to the final rectification still, which consists of a dephlegmating column standing on a boiler. The contents of this final still are distilled off by steam, the distillates, representing the finished motor spirit, passing away through a water-cooled condenser.

The process of rectification for the smaller gasworks as outlined above unfortunately embodies the necessity for the use of sulphuric acid and caustic soda. Small gasworks which undertake the manufacture of sulphate of ammonia have experienced most trying conditions with regard to the supply of acid during the past four years, but it must be remembered that since the armistice the demand for the product has fallen off so that many of the acid plants are now running on reduced power. There should, in fact, be a surfeit of sulphuric acid, so that no difficulty whatever need be experienced in obtaining supplies. Moreover, the amount required for the manufacture of motor spirit is almost trivial, being in normal cases not more than 3 per cent by volume of the crude spirit dealt with, or 5 per cent in abnormal instances where the original spirit may be of bad quality.

Cost of Recovery

The question of the cost of plant and cost of recovery is of chief importance, for upon it depends the ultimate price for which the spirit can be retailed to the public. It is, perhaps, in the direction of cost that the main difficulty of the smaller gasworks lies, in that if a standard price for benzol is fixed they will be at a disadvantage as compared with those works producing some hundreds of gallons per day. Although at the present time the average cost of rectified spirit is about 2s. 7d. at rectifiers' works, it is being sold by one large London gasworks at 2s. 4d. per gal. It is, however, an unfortunate fact that the cost of recovery of the crude spirit shows very wide differences when the various gasworks are considered. For instance, at one works the steam required for the purpose may be obtained from existing boilers with practically no extra expenditure, while at another it may be necessary to run a small supplementary boiler plant in connection with the process, this introducing the question of additional labor. Again, yet another establishment may be provided with a direct-fired still, requiring constant attention and stoking.

In these circumstances it is exceptionally difficult to give more than what could be considered an average figure as to cost of production, but for the normal works employing a plant of a capacity of 100 gal. per day, equivalent to 80 gal. of motor spirit, the cost of recovery and rectification may be taken as follows:

	£	s	d
100 gal. crude spirit at 1s. per gal.	5	0	0
Sulphuric acid at £6 per ton	0	4	0
Caustic soda at £25 per ton	0	1	3
Fuel, coke at 30s. per ton	0	0	8
Labor	0	10	0
Interest on capital at 6 per cent	0	2	9
Depreciation, wear and tear, at 7 per cent	0	3	3
	£6	1	11
Less 15 gal. of creosote at 3d.	0	3	9
Net cost of 80 gal. of motor spirit	£5	17	2

This represents a cost of 1s. 6d. per gal. for working expenses alone, and the cost of suitable cans, canning, etc., must be added. Thus it will be seen that, if anything approaching a reasonable profit is to be made, the present price of 2s. 4d. per gal. scarcely offers an alluring proposition to the gas undertakings, for a clear profit of approximately 15d. per gal. would be necessary to affect the receipts from by-products sufficiently to enable these undertakings to reduce the price of gas by 1d. per 1000 cu. ft.

A final word is necessary with regard to any standard specification of quality which may be prescribed for the spirit. Specific gravity undoubtedly will be an important feature of any such specification; but if this is restricted too closely, within certain limits it will probably disqualify much of the vertical retort benzol, containing paraffins such as hexane. These paraffins are in reality constituents which closely resemble gasoline, and are therefore not objectionable.—*The Times Engineering Supplement.*

PERSONAL NOTES OF THE MEMBERS

J. R. Archibald has accepted a position as retail sales manager with the Cadillac Motor Sales Co., Ltd. and is stationed at Winnipeg, Canada. He was formerly district representative of the Maxwell-Chalmers Motor Corporation and was stationed at Walkerville, Ont., Canada.

Paul E. Breneman has accepted the position of chief engineer of the all-steel body division with the C. R. Wilson Body Co., Detroit, Mich. For the past 5½ years he was chief draftsman with the Edward G. Budd Mfg. Co., Philadelphia, Pa.

Charles A. Cook has resigned as sales engineer of the Detroit Accessories Corporation, Detroit, Mich, to accept the position of chief engineer with the King Motor Car Co., also of that city.

R. H. Combs, past-councilor of the Society, who has been for several years general manager of the Prest-o-Lite Co. of Canada, Toronto, Ont., has been made general manager of the Canadian National Carbon Co., and will be located in that city. Both of these companies are enlarging extensively their facilities for production.

O. L. Curtis has resigned as engineer and sales manager of the Racine Mfg. Co., Racine, Wis., to accept a position with the Ligonier Auto Body Co., Ligonier, Ind.

R. E. Davis has been appointed production manager of the Square Turn Tractor Co., Norfolk, Neb. He was formerly tractor engineer with the Advance Rumely Co., Battle Creek, Mich.

Ralph W. Davis has resigned as chief engineer of the Mitchell Motors Co., Racine, Wis., a position which he has held for the past 2 years. He expects to remain in Racine and to form an automobile manufacturing company.

Stephen O. De Orlow has accepted a position in the research laboratory of the General Motors Corporation, Detroit, Mich.

I. Clayton Dickover has severed his connection with the Maccar Truck Co., Scranton, Pa., and is now at Wabash, Ind.

William A. Evans has severed his connection with the English & Mersick Co., New Haven, Conn., and has organized the E. V. B. Mfg. Co. for the manufacture of automobile body hardware in that city.

George B. Fuller has accepted a position in the truck department of the Packard Motor Car Co., Detroit, Mich. He was formerly assistant chief engineer of the Glenn L. Martin Co., Cleveland, Ohio.

J. M. Goldman has been appointed engineer of tests for the War Department and is now located at the Bureau of Mines experiment station, Pittsburgh, Pa. He is engaged in experiments and tests there and also at the Bureau of Standards, Washington, some of his work being along the lines of developing low-pressure oil burners for tilting aluminum and brass furnaces and thermostatically controlled oil burning devices for low-pressure steam and hot air heating.

Leigh M. Griffith, senior staff engineer of the National Advisory Committee for Aeronautics, has transferred his office from Washington to the research laboratory of the Committee at Langley Field, Hampton, Va.

C. E. Heckel, who was formerly a designer in the motor truck engineering department of the Pierce-Arrow Motor Car Co., Buffalo, N. Y., has accepted a position as designing engineer with the Holt Mfg. Co., Peoria, Ill.

George T. Homeier who was formerly with Dodge Bros., Detroit, Mich., is now general superintendent of the Lake Shore Engine Works, Marquette, Mich.

A. H. Hudson is now manager of purchases with the Isko Co., Chicago, Ill. He was formerly purchasing agent for the American International Steel Corporation, New York City.

F. A. Ingalls who was formerly president of the Ingalls-Shepard Forging Co., Harvey, Ill., has been appointed a vice-president of the Wyman-Gordon Co., Worcester, Mass., following the absorption by that organization of the Ingalls-Shepard Co. In the future this plant, which will be known as the Ingalls-Shepard division of the Wyman-Gordon Co., will be under the active management of Mr. Ingalls.

Rudolph H. Klauder has resigned as sales engineer with the National Carbon Co., Cleveland, Ohio, to accept the position of sales manager for the Bissinger Co., also of that city.

Frank B. Knepper who served with the American Expeditionary Force in France with the rank of second lieutenant in the Engineer Reserve Corps has been discharged from the Government service. He has accepted a position in the rate department of the Dayton Engineering Laboratories Co., Dayton, Ohio.

John Kralund was elected second vice-president in charge of production at a recent meeting of the board of directors of the Doehler Die-Casting Co., Brooklyn, N. Y. He was formerly factory manager of the company.

H. C. McIntyre has recently returned from Germany, where he was on duty in the Ordnance Department of the American Expeditionary Force with the rank of major. He has been appointed assistant chief engineer at the Rock Island Arsenal, Rock Island, Ill.

John H. McNamara has been elected vice-president of the Keuka Industries, Inc., Hammondsport, N. Y., which recently purchased the factories of the Curtiss Aeroplane & Motor Corporation at that city. He was formerly factory manager of the Curtiss plant.

K. B. MacDonald has been elected secretary and treasurer of the Keuka Industries, Inc., Hammondsport, N. Y. He was formerly production manager of the Buffalo plant of the Curtiss Aeroplane & Motor Corporation.

Harry Bowers Mingle who was formerly president of the Standard Aircraft Corporation, Elizabeth, N. J., has resumed the practice of law with offices in the Woolworth Building, New York City.

Charles Pack, chief chemist and metallurgist of the Doehler Die-Casting Co., Brooklyn, N. Y., was elected secretary and chief chemist at a recent meeting of the board of directors of that company.

E. F. Paepper has resigned as chief engineer of the All-American Truck Co., Chicago, Ill., to accept a position with the Superior Motor Truck Co., Atlanta, Ga.

Raymond C. Pollock has accepted a position as engine inspector with the Daniels Motor Car Co., Reading, Pa. He formerly held a similar position with the Standard Steel Car Co., Butler, Pa.

Ernest Rolland has accepted a position in the industrial engineering department of the Foundation Co., New York City.

Earl H. Seelbach has accepted a position as assistant engineer in the truck department of the H. H. Franklin Mfg. Co., Syracuse, N. Y. He was formerly chief draftsman in the same department of the Pierce-Arrow Motor Car Co., Buffalo, N. Y.

Arthur J. Slade who served as chief of the engineering branch of the Motor Transport Corps with the American Expeditionary Force with the rank of lieutenant-colonel has been elected president of the recently organized Motor Transport Post of the American Legion.

B. J. Steelman has been appointed vice-president of the Wanner Malleable Iron Co., Hammond, Ind. He formerly held the same office with its predecessor, the Hammond Malleable Iron Co. There have been no changes in the officers or management.

David R. Swinton who served with the American Expeditionary Force in the Motor Transport Corps with the rank of captain has been discharged from the service and has

accepted the position of sales engineer with the American Autoparts Co., Detroit, Mich.

P. A. Tanner has been appointed secretary of the Splitdorf Electrical Co., Newark, N. J., and will exercise supervision over the service, advertising and catalog compilation work. For the past 4 years he has been mechanical engineer, service and advertising manager of the Sumter division of the company with headquarters at Chicago, Ill.

Edward G. Topie has been appointed superintendent of tools with the Briscoe Motor Co., Jackson, Mich. He has been recently discharged from the Ordnance Department with the rank of first-lieutenant.

H. C. Wanner, treasurer and general manager of the Hammond Malleable Iron Co., Hammond, Ind., has been appointed to the same offices in the Wanner Malleable Iron Co., also of that city, which has been formed to take over the business of the former company, which now ceases to exist.

E. R. Waterman has resigned as sales manager with Edward V. Hartford, Inc., Jersey City, N. J., and has accepted a position in the New York City office of the Splitdorf Electrical Co., Newark, N. J.

Ernest H. Wehrli has accepted a position in the engineering department of the International Motor Co., New York City. He was formerly a designer for the Militor Corporation, Jersey City, N. J.

William G. Wood, Jr., has accepted a position with the Autocar Sales & Service Co., Inc., New York City, and is specializing in truck body design.

J. Zagora has resigned as designing and production engineer with the Anderson Motor Co., Rock Hill, S. C., and has organized the J. Zagora Mfg. Co., Charlotte, N. C., to manufacture automotive parts and various other smaller assemblies required in the automotive and its associated industries.

OBITUARIES

Leo G. Benoit, technical manager of the Tips Aero Motor Co., Inc., Woonsocket, R. I., died Jan. 3, 1920. He was born in that city, June 4, 1888, and received his education in the local schools, later graduating from a special course in mechanical drawing and mathematics. His practical experience included progressive stages of drafting and machine design with several different firms. In 1916, he became associated with the Tips company, which was incorporated at that time, in the development of a new 400-hp. aviation engine and had charge of its design and construction at the time of his death. He was elected to the Associate member grade in the Society, Dec. 10, 1917.

Theodore Douglas, consulting engineer and president of the Duplex Engine Governor Co., Brooklyn, N. Y., died Jan. 21, 1920, at his home in Scarborough, N. Y., after an illness of about 4 months. He was born March 2, 1868, at Washington, and following his preliminary schooling received his technical education at the Sheffield Scientific School of Yale University, from which he was graduated in 1894.

As a mining engineer he traveled extensively in Europe, Africa and Central America, as well as in the United States, spending also 3 years in archaeological research work in Italy. For 15 years Mr. Douglas was active as a chemist and metallurgical engineer, conducting also a consulting engineering practice involving many public utility problems relating to municipal railway transportation and designing various types of subway car. He was for some time president of the Hudson Equipment Co.

Mr. Douglas was the inventor of the duplex engine governor, a device for regulating gasoline engines and controlling the speed of motor trucks. For the past 7 years his activities were largely confined to making this an engineering and a commercial success. Numerous other inventions were, however, developed and perfected by him, the last being an automatic safety ignition interrupter device for airplanes. This was used as an ignition cut-off on the engine, the instrument acting as a safety appliance in case of propeller breakage or similar accidents. The Government was vitally interested in this during the war, and the Navy Department used it on different types of airplane after extensive tests. It is now being successfully used on commercial airplanes. He was also connected with many prominent engineering and educational societies throughout the country. He was elected to Member grade in the Society, Jan. 21, 1914.

In the demise of Theodore Douglas, the Society has suffered a distinct loss. As in his professional life, he made the bravest of fights during the several months of his most distressing and hopeless illness. He was a man of high ideals and of great preciseness of character. A most assiduous worker, he gave the industry the benefit of his unusual inventive ingenuity. The sincerity of effort of Mr. Douglas, as a man and as an engineer, will not soon be forgotten by the members.

Darwin S. Hatch, managing editor of *Motor Age*, died Jan. 20, 1920, at Chicago, Ill., from pneumonia. He was born Feb. 19, 1883, at Kentland, Ind. Following his high school education he prepared for entrance to the West Point Military Academy, but attended Purdue University instead, and was graduated in 1906, receiving the degree of bachelor of science in electrical engineering. After several years as maintenance and equipment foreman for the Chicago Telephone Co., he served the Chicago Aeronautic Supply Co. in the capacity of manager during 1909 and 1910. Becoming connected with the editorial staff of the Class Journal Co. in 1911 as a technical writer on automobile and kindred subjects, he advanced to the managing editorship of *Motor Age* and was thus actively engaged when stricken with the illness that terminated in his untimely decease. Mr. Hatch was elected to Member grade in the Society, Feb. 24, 1914.

The passing of Darwin S. Hatch, widely and most favorably known to the members, was a great shock and most deeply deplored. Mr. Hatch's effective and markedly conscientious work as editor of *Motor Age* is well recognized. A quiet and unassuming worker, he has much achievement to his credit. His service as Secretary of the Mid-West Section of the Society was conspicuous in faithfulness and resulted indirectly in benefit to the whole membership.

Robert Clarkson Reid, who died Jan. 11, 1920, was born at Kingston, N. Y. on July 8, 1875. He received his preparatory training in the public schools of Nyack, N. Y., and a technical education at Cooper Union, New York City. In 1892 he entered the shops of R. Hoe & Co., printing press manufacturers, and in 1895 became one of their draftsmen. For nearly 2 years following 1898 he was an engineering draftsman for the American Sugar Refining Co. and during a part of 1900 was an engineer for the Corn Products Co. He became New York City manager for the Chapman Valve Mfg. Co., Indian Orchard, Mass., in October, 1900. Since 1908 Mr. Reid had been a salesman for the Metzger Motor Car Co., Englewood, N. J. He had been a member of the American Society of Mechanical Engineers since 1907. He was elected to the Member grade in the Society April 28, 1911.

Applicants *Qualified*

The following applicants have qualified for admission to the Society between Dec. 27, 1919, and Jan. 24, 1920. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff.) Affiliate; (Aff. Rep.) Affiliate Representative; (E. S.) Enrolled Student; (S. M.) Service Member; (F. M.) Foreign Member.

Ackerman, Albert H. (M) 305 West Randolph Street, *Chicago, Ill.*

Adams, George H. (M) sales service engineer, Bock Bearing Co., Toledo, Ohio, (mail) 2029 West Fiftieth Street, *Cleveland, Ohio.*

Ames Holden Tire Co., Ltd., (Aff.) 1221 Mount Royal Avenue East, *Montreal, Canada.* Representatives: H. R. Brandt, technical superintendent; A. E. Parmenter, factory managei; W. B. Wiegand, director of manufacturing.

Arnold, Robert Melville (E.S.) student, Massachusetts Institute of Technology, Cambridge, Mass., (mail) 203 Bay State Road, *Boston, Mass.*

Barkley, Charles M. (M) consulting engineer, Box 222, *Racine, Wis.*

Bechtel, George Harry (M) consulting engineer, 4 Palmeria Avenue Mansions, *Hove, Sussex, England.*

Becker, William O. (M) mechanical engineer, Standard Parts Co., *Cleveland, Ohio,* (mail) 4008 Prospect Street.

Bennett, W. Burr (M) chief engineer and president, Wayne Engineering Co., Box 246, *Honesdale, Pa.*

Berquist, Olof C. (J) aeronautic draftsman, Naval Aircraft Factory, *Philadelphia, Pa.,* (mail) 3319 Walnut Street.

Bikle, William F. (M) design and construction engineer, Crawford Automobile Co., *Hagerstown, Md.,* (mail) Box 409.

Boivin, Lloyd C. (A) automobile electrician, DeBerry Brothers & Bennett, 228 Drumm Street, *San Francisco, Cal.,* (mail) 2550 Polk Street.

Brouthers, A. H. (A) automobile dealer, *Pittsfield, Mass.,* (mail) 32 Northumberland Road.

Butterworth, Henry L. (J) laboratory assistant, Nordyke & Marmon Co., *Indianapolis, Ind.,* (mail) 1706 North New Jersey Street.

Carr, Byron (A) superintendent, Overland Pacific Co., *Seattle, Wash.,* (mail) 1637 Twenty-first Avenue.

Cawthorne, George S. (M) chief engineer, Master Trucks, Inc., *Chicago, Ill.,* (mail) 4510 Greenview Avenue.

Coghlin Co., Ltd., B. J. (Aff.) 2050 Ontario Street East, *Montreal, Canada.* Representative: Bernard W. Coghlin, president.

Copland, Robert Y. (M) mechanical engineer, Dominion Tire Factory, 149 Strange Street, *Kitchener, Ont., Canada.*

Cosner, Richard E. (A) machine shop superintendent, G. A. Schacht Motor Truck Co., *Cincinnati, Ohio,* (mail) 66 De Camp Avenue.

Crane, Jasper E. (M) manager cellulose division, chemical department, E. I. du Pont de Nemours & Co., *Wilmington, Del.*

Davison, Ralph E. (J) assistant superintendent, Curtiss Engineering Corporation, Garden City, N. Y., (mail) 12 Ormond Street, *Hempstead, N. Y.*

Dawson, Frederick Y. (M) research engineer, Naval Aircraft Factory, *Philadelphia, Pa.,* (mail) 5719 Addison Street.

Dillard, Col. James B. (M) Ordnance Department, Sixth and B Streets, *Washington.*

Dishman, H. C. (M) sales engineer, Raybestos Co., *Bridgeport, Conn.*

Eade, Walter F. (A) aeronautical engineer, Massachusetts Institute of Technology, *Cambridge, Mass.*

Eastman, Robert L. (M) tractor department, Electric Wheel Co., *Quincy, Ill.*

Edscorn, G. E. (J) automotive engineer, Wagner Electric Mfg. Co., 6400 Plymouth Avenue, *St. Louis, Mo.*

Emmons, Harold H. (A) attorney, 1301 Ford Building, *Detroit, Mich.*

Flanders, Ralph E. (M) manager, Jones & Lamson Machine Co., *Springfield, Vt.,* (mail) 41 Pleasant Street.

Foster, Harry C. (A) master mechanic, Indiana Lamp Co., *Connersville, Ind.*

Gray, Jack (A) sales engineer, Champion Ignition Co., Flint, Mich., (mail) 1044 Third Street, *Detroit, Mich.*

Gwynne, George R. (A) manager, automotive department, Continental Oil Co., *Denver, Col.*

Harding, S. V. (M) directing engineer, Midwest Engine Co., Nineteenth and Martindale Streets, *Indianapolis, Ind.*

Harper, George W. (M) assistant chief engineer, Columbia Axle Co., *Cleveland, Ohio,* (mail) 850 East Seventy-second Street.

Harrington, W. F. (M) research engineer, Willys-Overland Co., *Toledo, Ohio.*

Harrison, R. E. (J) mechanical draftsman, Holt Mfg. Co., *Stockton, Cal.,* (mail) 402 Y. M. C. A.

Haskins, Earl A. (M) E. E. Springer Co., Twelfth Street and Monticello Avenue, *Norfolk, Va.*

Henry, Ferdinand G. (M) chief engineer, George White Co., Jersey City, N. J., (mail) 1854 Seventh Avenue, *New York City.*

Hill, G. Clifford (A) superintendent, Eldridge-Buick Co., 802 East Pike Street, *Seattle, Wash.*

Indiana Lamp Co. (Aff.) *Connersville, Ind.* Representatives: George W. Ansted, secretary and treasurer; Chester A. Craig, superintendent; Loren E. Glass, purchasing agent; Watt E. Ray, assistant secretary and treasurer; William F. Thoms, sales manager.

Johnson, Carl E. (J) mechanical draftsman, Domestic Engineering Co., *Dayton, Ohio,* (mail) 106 Baker Street.

Jones, J. J. (A) president and general manager, Jones Motor Car Co., *Wichita, Kan.*

Kaye, Le Roy Allan (E.S.) student, Armour Institute of Technology, *Chicago, Ill.,* (mail) 5422 Michigan Avenue.

Keeler, Raymond W. (J) draftsman and designing engineer, Kelly-Springfield Motor Truck Co., *Springfield, Ohio.*

Kegerreis, Claude S. (J) research engineer, Purdue University, Lafayette, Ind., (mail) 204 South Salisbury Street, *West Lafayette, Ind.*

Keppler, George (A) superintendent, G. A. Schacht Motor Truck Co., *Cincinnati, Ohio,* (mail) 1732 Republic Street.

Klingner, Adolf F. (M) general superintendent and mechanical engineer, Silvex Co., *South Bethlehem, Pa.*

Knowles, George F. (M) production manager Stephens Motors Works, Moline Plow Co., Inc., *Freeport, Ill.*

Kraemer, Emil (M) engineer, 236 Bay Tenth Street, *Brooklyn, N. Y.*

Langbein, H. W. (J) manager, American Gear & Transmission Co., *Los Angeles, Cal.,* (mail) 1629 Vineyard Street.

Larson, Jesse L. (A) factory superintendent, Oklahoma Auto Mfg. Co., *Okay, Okla.*

McGrath, Raymond D. (A) assistant sales manager, Federal Pressed Steel Co., *Milwaukee, Wis.,* (mail) University Club.

McManus, Albert Edward, Jr. (A) sales engineer, F. G. Ericson, *Toronto, Ont., Canada,* (mail) Mail Building.

Mainland, John (M) chief engineer, thresher works, Advance Rumely Co., *La Porte, Ind.,* (mail) 507 C Street.

Marcum, Arthur I. (J) draftsman, Holt Mfg. Co., *Stockton, Cal.,* (mail) 306 East Flora Street.

Mars, Walter E. (J) draftsman, Brewster & Co., Long Island City, N. Y., (mail) 204 West 121st Street, *New York City.*

Mattix, Paul R. (A) *Kokomo, Ind.*

Mertins, William R. (M) assistant chief engineer, Mitchell Motors Co., Inc., *Racine, Wis.,* (mail) 1505 Flett Avenue.

Miller, Major Reuben, Jr. (S.M.) associate chief, engineering division, Motor Transport Corps, *Washington,* (mail) Wardman Park Inn.

Morehead, William C. (M) president, Great Lakes Boat Building Corporation, 333 Becher Street, *Milwaukee, Wis.*

Muller, Otto (J) mechanical draftsman, International Motor Co., *New York City,* (mail) 588 Van Nest Avenue.

Naramore, H. B. (A) secretary, Bridgeport Coach Lace Co., *Bridgeport, Conn.,* (mail) 805 Wood Avenue.

Newcomb, Benjamin R. (A) patent attorney, 211 Victor Building, *Washington.*

Nides, Emanuel (M) chief engineer, Standard Steel & Bearings, Inc., *Philadelphia, Pa.,* (mail) 629 North Seventh Street.

Parsons, Ben Gillespie (M) chief of engineering, Dayton Wire Wheel Co., *Dayton, Ohio.*

Porter, Robert (M) vice-president, motor equipment division, Jaxon Steel Products Co., *Detroit, Mich.,* (mail) Box 1, North End Post Office.

Potter, M. R. (A) service manager, Genesee Motor Car Co., *Syracuse, N. Y.,* (mail) 610 West Onondaga Street.

Pratt, William Holley (A) sales engineer, motor equipment division, Klaxon Co., 3066 West Grand Boulevard, *Detroit, Mich.*

Ressegiue, O. H. (A) European sales manager, Hupp Motor Car Corporation, Detroit, Mich., (mail) 11 Rue Scribe, *Paris, France.*

Robinson, I. R. (A) photographic section head, Naval Aircraft Factory, *Philadelphia, Pa.,* (mail) 322 Mercy Street.

Rothaug, Walter H. (A) instructor, College of the City of New York, New York City; production engineer, Standard Aircraft Corporation, Elizabeth, N. J., (mail) *Short Hills, N. J.*

Schwager, H. A. (M) designing engineer, Ordnance Experimental Shop, Detroit, Mich., (mail) 510 West Fourth Street, *Royal Oak, Mich.*

Sears, Lester M. (M) vice-president and general manager, Towmotor Co., Bliss Road, Euclid Village, *Cleveland, Ohio.*

Sevin, Robert E. (J) engineer, Maxwell Motor Co., Inc., *Detroit, Mich.,* (mail) Y. M. C. A. Building, Adams Avenue.

Shuart, Arthur C. (J) layout draftsman, Reed & Glaser, *Indianapolis, Ind.,* (mail) 224 East Pratt Street.

Siddeley, E. H. (J) assistant works manager, Siddeley-Deasy Motor Car Co., Ltd., *Coventry, England.*

Steinfurth, Louis, Jr. (A) checker, engine department, Cleveland Automobile Co., Cleveland, Ohio, (mail) 13714 Sixth Avenue, *East Cleveland, Ohio.*

TALBOT, JAMES M. (A) sales manager, flexible shaft department, S. S. White Dental Mfg. Co., 7 Union Square, *New York City.*

TANNER, WILLIAM RAYMOND (A) assistant divisional superintendent in charge of assemblies, International Harvester Co., *Akron, Ohio,* (mail) 11 Arch Street.

TOMPKINS, J. E. (A) service manager, Couch-Haas Co., Inc., Brooklyn, N. Y., (mail) 314 Barclay Street, *Flushing, N. Y.*

TRUESDELL, FRED ADRIEN (M) assistant engineer and chief draftsman U. S. experimental tractor department, Van Dorn Iron Works, *Cleveland, Ohio,* (mail) 13604 Glenside Road.

TRUSCOTT, STARR (M) assistant in lighter-than-air department, Bureau of Construction and Repair, Navy Department, *Washington.*

VAN VLIET, JOHN D. (M) chief engineer, Cantilever Aero Co., New York City, (mail) *Amityville, N. Y.*

WALKER, EDWIN C. (M) designing engineer, Standard Parts Co., Eleventh and Walnut Streets, *Cleveland, Ohio.*

WARD, LOUIS M. (M) secretary and manager, Cushman Motor Works, *Lincoln, Neb.*

WARNER, C. B. (M) chief engineer, Nelson Motor Truck Co., *Saginaw, Mich.*

WEYL, PIERCE ALBERT (E.S.) college of engineering and architecture, University of Michigan, *Ann Arbor, Mich.,* (mail) 427 Thompson Street.

WHISLER, T. CLIFFORD (J) engineering department, General Ordnance Co., *Derby, Conn.*

WILLIAMS, CHARLES LESLIE (M) designer, Samson Tractor Co., *Janesville, Wis.*

WOOLSON, LIONEL M. (M) engineer of tests, Packard Motor Car Co., *Detroit, Mich.,* (mail) 900 Burns Avenue.

YELM, C. W. (M) general manager, mechanical goods division and consulting engineer, Gates Rubber Co., *Denver, Col.,* (mail) 2070 Cherry Street.

YOUNG, C. G. (M) John N. Willys Export Corporation, 165 Broadway, *New York City.*

Applicants *for* Membership

The applications for membership received between Dec. 27, 1919 and Jan. 24, 1920, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

AYR, DAVID, general manager, Machine Stamping Co., Ltd., *Toronto, Ont., Canada.*

BELL, ERNEST ALFRED, general manager and chief engineer, Globe Motor & Taxi Co., Queens Bridge Street, *South Melbourne, Australia.*

BOWLER, CHARLES LEE, research engineer, Locomobile Co. of America, *Bridgeport, Conn.*

BRANIES, OTTO M., mechanical engineer, Dayton Fan & Motor Co., *Dayton, Ohio.*

BRAUTIGAM, H. H., secretary and manager, Bridgeport Motor Co., *Bridgeport, Conn.*

BUREAU, ACHILLE G., engineering department, Edward G. Budd Mfg. Co., Twenty-fifth Street and Hunting Park Avenue, *Philadelphia, Pa.*

CARLSON, ERNEST F., draftsman, tractor design, Hart-Parr Co., *Charles City, Iowa.*

CHAMBERS, CLINARD F., purchasing, Texas Co., 17 Battery Place, *New York City.*

CHARLES, RICHARD E., stock supervisor, Buick Motor Co., 630 West Fifty-second Street, *New York City.*

CHULSTROM, JOHN, mechanical draftsman, Class Journal Publishing Co., 239 West Thirty-ninth Street, *New York City.*

COX, CLYDE CURTIS, chief engineer, Jackson Motors Corporation, *Jackson, Mich.*

COX, NEWTON, manager, motor vehicle department, Sinclair Refining Co., *Chicago, Ill.*

CRAWFORD, JOHN X., automobile mechanician, Clark & Kendrick, Inc., 143 West Fifty-first Street, *New York City.*

DICK, WILLIAM F., layout draftsman, W. W. Shaw, 310 East Huron Street, *Chicago, Ill.*

DENNEEN, F. S., manager motor truck division, Grant Motor Car Corporation, *Cleveland, Ohio.*

DISSETTE, JOHN W., chief aircraft section, director of sales office, 2513 Munitions Building, *Washington.*

DOOLITTLE, ALBERT H., general manager, Claudel Carburetor Co., 192 Jackson Avenue, *Long Island City, N. Y.*

GIBBS, CARL C., sales manager, National Malleable Castings Co., *Cleveland, Ohio.*

GILL, HARRY H., chief engineer, Cummings Machine Co., *Minster, Ohio.*

GORHAM, WILLIAM R., mechanical engineer, Central Motor Mfg. Co., *Tokyo, Japan.*

GRINER, OTIS EDWARD, sales and service engineer, Midwest Engine Co., *Indianapolis, Ind.*

HALL, PAUL BISHOP, chief engineer and sales manager, American Die Castings Co., *Indianapolis, Ind.*

HAMILL, W. A., assistant sales manager, Neel Cadillac Co., *Philadelphia, Pa.*

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